Heating · Piping and Air Conditioning

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Proper Control of Air Circulation Important in Heating and Ventilating

By Lee P. Hynes*

IN designing a satisfactory heating and ventilating system for a room or building, it is necessary to make a careful study of many factors besides the actual size, shape, and heat losses of the structure. It is very essential to consider the internal air movements which are likely to occur during normal or unusual operating conditions, and to control them. Many large rooms having heated floors above and below lose heat principally through external walls and window surfaces and therefore do not need much heat applied to interior spaces. Large rooms having a roof directly over them, particularly where skylights are used, require heat distribution to overcome the cold downward movement of air in the interior as well as along the walls. Rooms with high ceilings must be treated differently from those of normal height.

Rooms having large exterior openings or doors which are frequently opened present difficult heating problems due to the great inrush of cold air along the floors every time a door is opened. There is also a considerable infiltration of cold air at such doors while closed. In some shops and factories a large amount of outdoor air must be introduced to overcome contamination from various sources and this likewise introduces heating and ventilating problems.

Each job must be carefully studied and suitable equipment provided to take care of the special operating conditions under which the space will be used. Mechanical circulation of the air is usually necessary in order to give adequate control.

Garages and Bus Terminals

Garages, bus terminals, repair shops, automobile assembly plants, dye houses, electric substations, restaurants, and paint rooms offer examples which require

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Fig. 1—Old street car barn converted to bus terminal, with combination heating and ventilating units, and exhaust fans at floor level

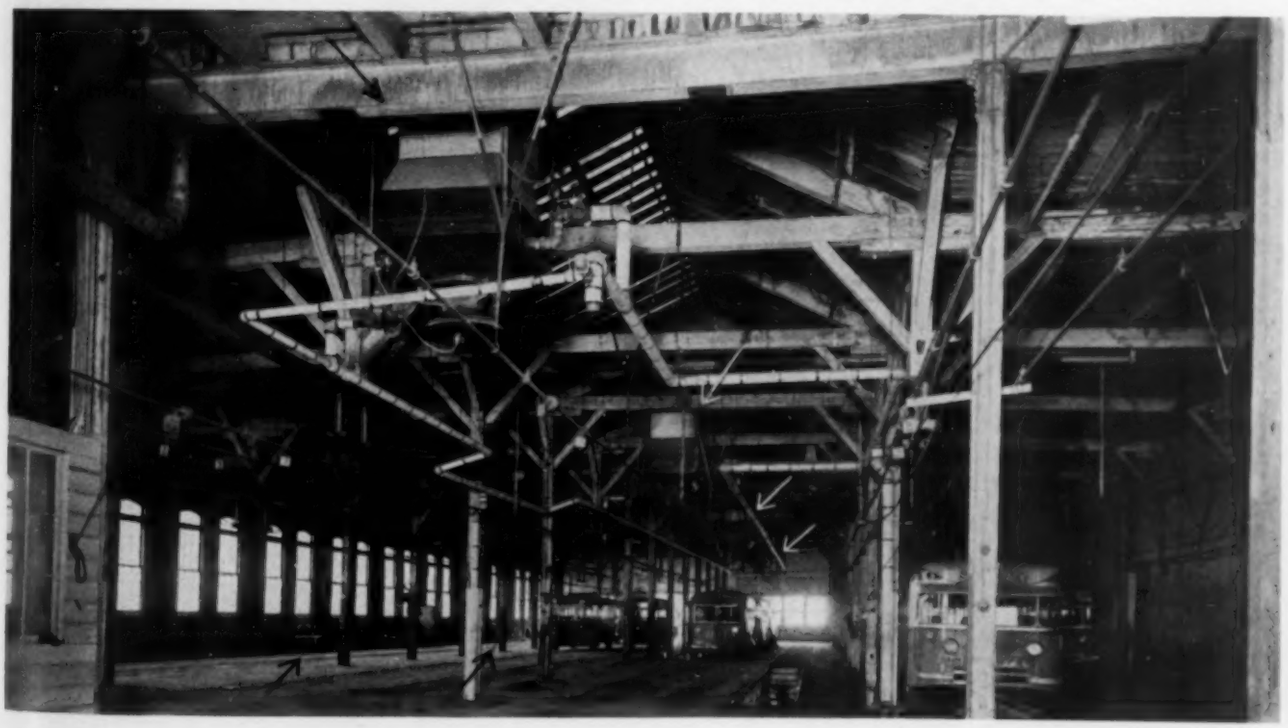


Fig. 2—All electric units heat this bus terminal, where steam for heating was not available

careful individual design. Garages and bus terminals often have low ceilings and always have large doors which are frequently opened. They have concrete floors along which incoming cold air flows and causes discomfort, particularly when men have to lie on the floor while making inspections or repairs.

Carbon monoxide gas and other products of combustion given off from engine exhausts tend to lie along the floor and become a serious hazard to men working there. In warm weather natural ventilation may be ample if large openings are provided in opposite walls near the floor level so that large volumes of outdoor air can flow across and sweep away the gases. This should also be supplemented with ample overhead vents to increase air circulation.

A desirable measure of dilution is one part of carbon monoxide gas in 10,000 parts of air. As one idling car engine may produce as much as 40 cu ft per hr of monoxide gas, it is obvious that very large volumes of outdoor air may be necessary under all weather conditions. In cold weather natural ventilation cannot be relied on because of cold drafts and excessive heating costs.

A well planned mechanical system can do the job much better and more economically than natural ventilation and the best plan is to bring in outdoor air at the roof. It should pass through suitable heaters and be driven down to the floor level at central points in the garage areas. This downflow of warm air must be designed with adequate linear velocity to reach the floor. After impinging there it rolls along under the cars toward the walls. Failure will result from using equipment with too low a nozzle velocity.

Suitable exhaust fans should be located in the walls to pull the gas laden floor stratum of air to them and discharge it outdoors. By a suitable arrangement of dampers, some air can be recirculated through the heaters and at times when no engines are running the system can be operated with complete recirculation and the exhaust fans can be shut down to conserve heat. Remote control and electrical interlocking permit complete flexibility to meet any operating condition.

Fig. 1 is a good illustration of how a complete change in operating conditions can be met by installing adequate mechanical equipment. This shows an old street car barn in Schenectady, N. Y., after being converted into a bus terminal. Natural ventilation had been ample in the days of trolley cars, but could not meet the new problems caused by bus engine exhaust gases. Buses need warming up before starting out on their runs and this demands a sudden peak in ventilating capacity and also in heating capacity during cold weather.

This problem was solved by installing overhead special combination heating and ventilating units arranged to bring in outdoor air through the roof, warm it, and drive it down to the floor at a proper velocity. Automatic damper arrangements control the outdoor air intakes and permit recirculation of any desired part of the air handled by the blower units. Powerful exhaust fans in the outside walls near the floor level draw the heavy, gas laden air from the floor level and discharge it outside. This arrangement provides an adequate supply of outdoor air and thoroughly warms it before it enters the working zone. This warm, outdoor air is forced down to the floor level from above, displacing the unwholesome fumes and the colder, heavier air at the bottom which is forced over toward the exhaust fans where it is discharged out of doors. Because of recirculation, the pocketing of warm air at the ceiling by convection is also prevented, which results in a greater economy of steam.

In the case of a new motor bus terminal of the Montreal Quebec Tramways Company with large doors

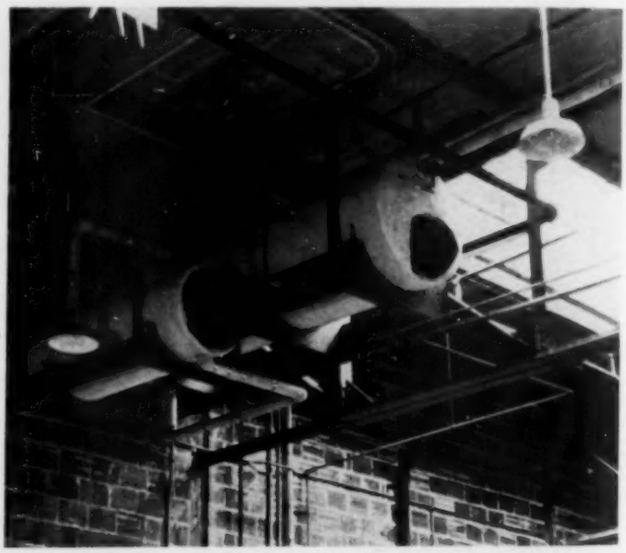


Fig. 3—Water is heated electrically for washing buses and for domestic use, the equipment being installed overhead

steam is not available and electricity is used as a heating medium. Special fan circulation unit heaters are suspended overhead from the structural roof trusses, as shown in Fig. 2. These heaters discharge warm air downward with a proper velocity to reach the floor level where it rolls along under the buses, making a comfortable working zone. Similar units in front of the doorways temper the incoming cold air. This system has operated very successfully under the severe Canadian winter conditions. This building is much smaller than the Schenectady car barn and it was possible to arrange a satisfactory system of natural ventilation supplemented by mechanical means.

This building also presented a special water heating problem because no steam was available. Large, overhead, electrically heated tanks were installed at various points as shown in Fig. 3. These supply hot water for washing buses and for toilet purposes in the repair shop and in the public waiting rooms. These waiting rooms are also heated by electricity with special fan circulation units. Both garage and waiting rooms have automatic temperature control.

Cold Drafts at Doors

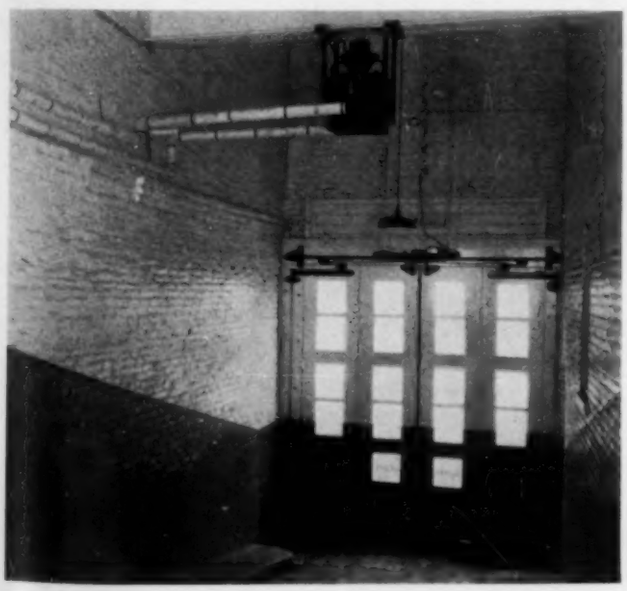
Every large garage is faced with the problem of cold drafts every time the doors are opened to let cars in or out. Wind velocities are seldom less than 5 mph and often are very much higher. Even at times of little wind the heavier cold air rushes in along the floor and displaces the lighter warm air which is forced out the upper half of the doorways.

Fig. 4 shows how this can be overcome. It illustrates one of the entrances to a large public garage in St. Paul, Minn., where cold drafts were cured by installing a high discharge velocity unit heater overhead just inside each door. A curtain of warm air is forced down to combat the inflow of cold air. Such air as does enter the main area of the garage is well tempered. It is then further distributed by similar heaters which constantly drive warm air down to the floor levels.

In a large, modern, auto assembly plant conventional methods of heating were tried but serious trouble was experienced because of the large openings required for the outgoing line of assembled cars. Actual tests showed that in cold weather, even when very little wind was blowing, the incoming draft of cold air had a velocity up to 400 fpm, and this cold draft could be felt several hundred feet away from the doorways.

As these doors must be kept open because of the constant outgo of finished product, it was necessary to redesign the heating system. After careful study the trouble was finally corrected by installing large, high velocity, unit air heaters near the ceiling in front of the open doorways. These drive warm air downward to the floor, forming a curtain against which the incoming cold air impinges, thus mixing and tempering it effectively.

Fig. 4—This unit heater in a public garage kills cold door drafts



Odors, Fumes and Condensation

Rooms for other uses require different handling. Restaurants face a special problem in overcoming odors and steam from the kitchens. This must be accomplished by exhausting a large volume of air from the kitchens with a consequent air flow from the dining rooms toward the kitchens. This, of course, requires the introduction of an adequate supply of outdoor, warmed air into the dining rooms without causing complaint of drafts. The frequent opening of outside entrance doors complicates matters and often the inrush of cold air can be checked by planned curtains of warm air forced across the doorways.

Paint rooms and other locations where light volatiles are evaporated can best be ventilated by discharging air at the ceiling by natural or mechanical means. In dye houses fog prevention can be accomplished by powerful exhausters located over the vats to carry away the vapors before they can condense.

In automatic electric power and telephone substations an unusual problem is encountered. It is unnecessary to provide for human comfort as no attendants are normally present. There may be short periods when heat is wanted while inspection or repairs are being carried on, but normally the only heat needed is for the prevention of condensation on electrical equipment. This can cause serious trouble through the breakdown of insulating materials.

Condensation indoors is caused just as dew is formed out of doors, by warm humid air coming in contact with cooler surfaces and being reduced to its dew point temperature. Ventilation alone will not overcome this trouble, but may even increase it. Ventilation does no harm during a period of falling outdoor temperature because any air which enters the building is warmed somewhat, hence its dew point is not reached.

The trouble comes after a cold spell. The interior walls and the equipment have been chilled. With a rising outdoor temperature, the air has an increased capacity to carry moisture and as the warm, humid air enters the building and meets the colder surfaces it is chilled to its dew point and condensation forms. The greater the volume of air introduced, the greater the amount of dew that forms inside the building.

One way to prevent condensation is to provide just enough heat to avoid reaching the dew point temperature of the incoming air and also to maintain a rapid recirculation of the internal air. Neither will do the job alone but by a properly proportioned combination condensation troubles can be avoided without wasting heat.

A well proved method is to install special fan circulation electric air heating units with an electric dew point control regulator. This does not keep the internal air temperature constant as a thermostat would do, but automatically causes it to rise and fall in harmony with outdoor temperature changes. By this means no heat is wasted. Exactly enough is called for to maintain the internal temperature just above the dew point temperature of the air which filters in from out of doors. This is all that is necessary because no condensation can form in the building unless the incoming air drops in temperature after entering.

[Figs. 1 and 4 courtesy L. J. Wing Mfg. Co. Figs. 2 and 3 courtesy E. W. Playford Ltd.]

Control of Air-Borne Organisms

Recent Studies on Bacteria in Air Reviewed

By H. C. Murphy*

Some of the recent studies on air-borne organisms are reviewed briefly, and the advantages of filtering air for controlling bacteria are suggested. . . . The author, who has had a wide experience in the control of bacteria by air filtration in different industrial processes, will discuss some of them in a later article

MANKIND lives and dies surrounded by myriads of invisible but living organisms known as bacteria. These single cell organisms, too small to be seen by the naked eye, have a profound influence on our daily lives. The average diameter of the spherical shaped bacteria (*cocci*) is perhaps 1 micron (approximately 0.00004 in.). Some idea of their size can be had by considering that 700 of these bacteria placed in a straight line end to end would pass through an ordinary pin hole.

Bacteria propagate by subdivision and it has been calculated that under ideal conditions the progeny of one bacterium would in five days fill a space equal to all the seas on earth. Nature has, of course, set up checks to prevent any such development, but the rapid multiplication of bacteria presents a real problem to the medical world. Early Sanskrit records indicate that the Aryan doctors 2,000 years ago recognized the spread of disease by contagion and were acquainted with the value of sterilization by heat in the control of infections.

Bacteria in Air

For more than a generation physicians have been taught that germ carrying droplets ejected when we cough, sneeze and talk, drop to the floor more or less like grains of sand, there to lie and die. Recent work at the Harvard School of Public Health by Wells and Stone has shattered this comforting doctrine. It was Flügge, in the latter part of the 19th century, who thought he had demonstrated that infectious droplets fall harmlessly to the ground and this had been quite generally believed until Wells and Stone so conclusively demonstrated the fallacy of this theory.

The organisms tested at Harvard were sprayed in distilled water into a sterile, microbe-tight chamber. Most of the droplets were smaller than a thousandth of an inch in diameter. They evaporated before they fell to the floor. Samples of air taken from the chamber at intervals of half an hour to eight days, however, disclosed thriving colonies of bacteria which could be seen with the naked eye. Some bacteria were thus carried about alive for days. Others died in less than an hour. The studies indicated that bacteria expelled into the air, and therefore easily inhaled, lived longer than those carried by liquids and foods. It would appear possible that the organisms responsible for respiratory infections are especially adapted for air transmission. At any rate, these bacteria from the throat and nose live long enough in air to transmit disease by inhalation.

These observations were not altogether unexpected, as there had been much previous information on this subject. Dr. A. I. Kendall, former dean, Northwestern University medical school, for instance, in his work "Civilization and the Microbe," published in 1923, demonstrates the dispersion of droplets and air-borne organisms in the following manner:

A harmless microbe, known as *bacillus prodigiosus*, is used as the test germ; it grows readily upon many organic substances, and is sometimes found in the air. A small amount is introduced into the mouth of the demonstrator, who takes his stand at one end of a room previously prepared for the experiment by exposing at various levels and places dishes of sterile gelatin. If the room is shown by previous experience to contain no organisms capable of growing as red colored spots on the gelatin, it may be assumed that the appearance of red growths on the gelatin plates, after a certain period of time, represents the bacteria that have actually transmitted through the air when the experimenter talks, coughs or sneezes. The results of many trials have shown quite definitely that bacteria are actually expelled from the mouth of an individual. Also, it appears, they may pass through the air and lodge in the nose or mouth of another individual.

In the experiments with *bacillus prodigiosus*, growths were obtained in the uttermost corner of the room and 40 ft distant, and in one instance the microbes were driven by a gentle current of air from the experimenter's room through a hall and up a flight of several stairs. Dr. Kendall states that the microbe leaving the lungs of a careless consumptive is sent upward and outward with thousands of its fellows, and scattered in the air about. Droplets very minute but large in comparison with the microbes, may remain suspended in the air for many feet or yards. In theaters, cars and crowded schools, the chances of inhaling them are very numerous.

Studies Made in New York City

Investigations in New York City by two groups of investigators were reported recently in the *American Journal of Public Health*. Sol Pincus, deputy commissioner of the New York department of health, and Arthur C. Stern, superintendent of the WPA air pollution survey, gave the following figures in the April, 1937, issue:

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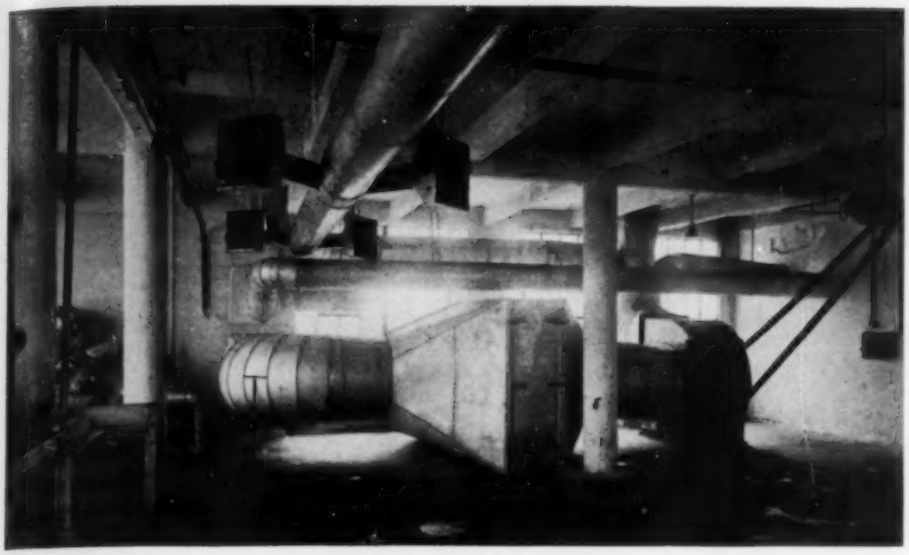

Bacteria control in the manufacture of nut butter. Air is first filtered in the plenum chamber shown in center of picture and refiltered at outlets of the ducts into drying room

Bacteria Per Cubic Foot of Air at Various Locations
Incubated 18 hours at 37 C
Week ending Friday, January 10, to week ending Friday, June 26, 1936

Type of Location	Sheep's Blood Beef Infusion Agar		Beef Infusion Agar	
	No. of Samples	Bacteria per Cu Ft of Air	No. of Samples	Bacteria per Cu Ft of Air
Schools	707	29.6	678	21.1
Subway	290	19.2	294	12.4
Theater—Non-ventilated (auditorium)	104	13.2	110	8.4
Theater—Ventilated (ducts)	149	3.1	114	1.4
Streets	143	11.2	134	8.5
Park	13	3.0	18	1.8

Streptococci per Cubic Foot of Air at Various Locations
(Sheep's Blood Beef Infusion Agar)
Sixteen weeks period—January 23 to May 15, 1936

Type of Location	No. of Samples	All Strepto.	Beta Hemolytic Strepto.	Alpha Hemolytic Strepto.
Schools	553	0.20	0.01	0.18
Subway	225	0.10	0.0003	0.085
Theaters—Non-ventilated (Auditorium)	78	0.04	0.001	0.038
Theaters—Ventilated (ducts)	116	0.03	0.0005	0.026
Streets	123	0.05	0.0001	0.045

From these tables it appears that the air in ventilated theaters studied in this survey has a smaller bacteria count than that in the schools or subways. In fact, the figures indicate a slightly lower bacteria count in the ventilated spaces than in the public parks. The ventilated theaters covered by these studies were equipped with air filters, while the schools were not so equipped. These filters were installed to eliminate dust and the results may be a further indication of a close connection between atmospheric dust and air-borne organisms.

Pincus and Stern state "While numerous studies of bacteria in air were made during the 19th century, the results were of negligible significance. However, during the first two decades of the present century, a number of workers in public health in the United States (Winslow, Baskerville, Rettger, Weinzirl, Soper, Browne, Ruehle, Whipple and others) using various modifications of the double sand filter originally designed by Petri and, in one instance (Rettger) using an improved water aeroscope, made studies of air in schools, streets, subways, dairies, and other places. Most of their efforts were concerned with the estimation of the number of bacteria per unit volume of air, and in some instances, the number of streptococci that appeared on litmus lactose plates. The wide variations in results leads one to suspect unavoidable errors in technic. Most of the methods demanded a separate piece of apparatus for each step in the process of isolating bacteria; that is, one apparatus to create air flow, another to measure it, still another to remove bacteria from the air, and a final one to grow them for counting.

"No doubt the complications entailed, combined with cumbersomeness of apparatus, hampered routine collection of samples. This may account for the waning interest in air analysis and its lapse into disfavor. The study of air bacteriology was greatly stimulated, if not actually revived, by the development of the Wells air centrifuge, in 1933. This centrifuge creates an air flow, measures the flow, precipitates the bacteria onto a thin layer of agar, and then permits of their being counted after incubation of these media on the same bottle.

"The ease with which samples are obtained by the Wells centrifuge simplifies the planning of routine collection of samples and allows the maintenance of fixed schedules for sampling. For example, six public schools have each been sampled twice a week on the same days of the week, in the identical rooms, and at the same hours of the day, for a period of approximately six months (until the schools closed for the summer vacation). Similarly, air in subways has been studied at the same hours five days a week. The same routine was followed in theaters, both before and during performances, and at the remaining types of locations.

"The table also gives the estimated number of streptococci per cubic foot of air in the same types of locations for a 16 week period from January 23 to May 15. It is seen that, here again, the schools have more streptococci than do any other type of location, and the subway is again second. The streets, in contrast with the findings for all bacteria per cubic foot, have slightly more streptococci than do the auditoriums of non-ventilated theaters. The ducts of ventilated theaters again have the lowest number. The division of the streptococci into Alpha and Beta streptococci show that the Alpha streptococci are in great preponderance. It is of very great interest that approximately 90 per cent of the Alpha hemolytic streptococci from schools and subway air are organisms whose normal habitat presumably is the human naso-pharynx, as determined by the Holman system of classification for streptococci."

[Concluded on p. 235]

Refrigeration Trouble-Shooting

By W. A. Stains*

ORDINARILY, the indications of trouble on a direct expansion coil air conditioning job of the usual type are low suction pressures; high suction pressures; hunting, cycling or wavering suction pressure gages; sweating or frosting suction lines; cold, sweating or frosting crankcases and cylinders; cold discharge lines; high air temperatures; and allied and related effects.

Wherever these faults are observed there are one or more causes. Low suction pressure may be due to several reasons, some of which are too small an evaporator, too light a load on a given evaporator, low refrigerant charge, inadequately sized liquid line, obstructions in liquid line, evaporator situated at considerable height above liquid receiver, and others less commonly understood.

High suction pressures indicate unbalanced conditions between the evaporator and compressor wherein the evaporator is absorbing more heat load than the compressor and condenser can disseminate. This may be due to unexpected high load conditions from the volume or temperatures of air passed through the evaporator and oversized expansion valves introducing too great a volume of liquid refrigerant into the evaporator.[1]

High head pressures at the valve may also produce high suction pressures or variable suction pressures because the valve, being sized for normal pressure drop across it, will allow more liquid to enter the evaporator, which increases the gas volume and the suction pressure rises accordingly. This effect is the same as an oversized valve and usually produces a cycling effect in the suction system comparable to a "flood through and pump down" cycle.

The oversized valve, or the correctly sized valve subjected to extremely high head pressures, produces a condition with thermostatically controlled expansion valves that seems paradoxical. This is because the flow of liquid through the valve into the evaporator is at a greater rate than the capacity of the evaporator to evaporate the liquid. Consequently some of the liquid passes through the evaporator to the location of the thermal bulb and due to the low temperature at the bulb closes the valve and stops the flow of liquid to the evaporator. The liquid in the evaporator is still flowing to the thermal bulb location and the time required for all the liquid to evaporate or flow past the bulb is the time the valve is closed. The complete evaporation of liquid quite likely is taking place at the entering end of the evaporator while a saturation still exists at the bulb; hence the valve remains closed until all the liquid is evaporated in the coil or tubes and a degree of superheat is set up again at the bulb to open the valve—and the same cycle recurs. This action reflects on the suction gage by wavering of the indicating hand.

Sweating and frosting lines can be attributed to several causes. Periodic sweating at high suction and frosting at low suctions indicates liquid feeding improperly through the valve. This may be due to improper size of the valve, improper application of thermal bulb, loose scale in the valve tending to hold open the valve too long, valve not sensitive enough to anticipate load changes, and combinations of these. It must be kept in mind that suction lines will sweat when everything is functioning perfectly if the lines are situated where the dew point temperature of the surrounding air is higher than the temperature of the suction gas, even though the suction gas is superheated. Frosted suction lines on air conditioning work are accompanied by low suction pressures and the reasons given for low suction pressures apply.

Cold, sweating or frosted crankcases are the results of more refrigerant in the evaporator than can be evaporated, which causes liquid to flow back to the compressor—termed "flooding through." Quite often, flooding through is a result of poor refrigerant distribution in the various tube passes of the evaporator; in other words, the liquid refrigerant is not fed to the several tube circuits in proportion to the load on them. This condition has caused many systems or parts of the system to be condemned; quite often the thermal valve is condemned when the trouble really lies elsewhere.

A cold compressor discharge pipe is an indication of refrigerant liquid flooding back to the compressor suction connection in considerable quantities. Usually a cold or sweating crankcase is accompanied by a cold compressor discharge line, but not always.

High air temperatures leaving the evaporator indicate, in the case of adequately sized evaporators, improper distribution of refrigerant through the evaporator; too little liquid supplied to the evaporator by too small a valve; a properly sized valve with the remote thermostatic bulb not properly located; shortage of refrigerant in the system, or insufficient supply of liquid to the valve due to restrictions in the liquid line; too small a liquid line; too small a king valve; or insufficient pressure at the valve. This latter condition may be due to an extreme vertical head between the receiver and the valve, which causes a drop in pressure at the higher point and forms gas which causes gas binding of the liquid. High air temperatures are also due to inadequately sized suction lines or insufficient displacement in the compressor. There are other reasons for high air temperature, such as air leakage between fan and coils on "draw through" systems where warm air is mixed with the cool air after leaving the coil before entering the fan. Many other reasons might be given, but as we are primarily concerned with the refrigeration phase of air conditioning they are neglected here.

Where the evaporator is not of sufficient capacity for the load the cause of high air temperatures is clearly evident.

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[1] William Goodman's article "Simple Way to Analyze Performance of Compressor-Evaporator Combinations," which was published in December, is of related interest.

In amplifying the foregoing causes, more explicit examples will be taken up, analyzed and explained with suggested remedies.

When too small an evaporator is used but the compressor capacity is sufficient, it is quickly evident that low suction pressures will ensue. In analyzing this, it becomes evident that there is a certain heat load that must be transmitted from the air passing through the evaporator to the surface of the evaporator and in turn to the refrigerant in the tubes. The refrigerant does its greatest work or absorbs the greatest number of heat units when it evaporates from a liquid to a gas. This capacity for absorbing heat in the transition from liquid to gas is the latent heat capacity or "value" of the refrigerant.

In changing from liquid to gas the volume of the refrigerant increases materially, as in the case of "Freon" at 37 lb per sq in. when 1 cu in. of liquid expands to 68 cu in. of saturated gas. In expanding this cubic inch of liquid into saturated gas approximately 3¼ Btu is absorbed. It becomes necessary therefore to expand as many cubic inches of liquid into a gas as there are 3¼ Btu in the total load. If the evaporator can evaporate only 0.8 or 0.9 cu in. of liquid for each 3¼ Btu that must be absorbed, it is self-evident that with the evaporator working at full capacity there just isn't enough surface or evaporator to handle the full load. Consequently the suction pressure drops, because the displacement of the compressor is greater than the amount of gas formed in a given time, and as not enough Btu's have been taken out of the air, the air temperature is higher than desired. In this case the remedy is a larger evaporator, precooling of the air by some auxiliary means, or lessening the total heat load of the air being conditioned by insulation or elimination of heat producing means. Necessarily it will require slowing up the compressor to balance the actual heat transfer, or even lower suction pressures may be encountered.

When the compressor and evaporator are properly sized and balanced, pipe lines are properly sized, valves are correctly fitted and applied, sufficient refrigerant is in the system, and everything checks out correctly, yet the suction pressure is too low, the trouble will be found in the air loading; the quantity of air passed through the evaporator is too small or the inlet dry bulb or wet bulb temperatures or both are too low. The air temperatures and volume should be checked to determine if they are in accordance with the design conditions.

If the air volume is too low it may be due to unexpected or undue resistance in the system, which may or may not be easily corrected. Testing with a manometer or pitot tube will indicate if such is the case. If resistance is in accordance with design, the fan delivery is apparently short in volume due to the fan speed being too slow. The fan should be speeded up, but bear in mind that if the air volume is low but resistance is right, the increased air volume secured by greater rpm of the fan will increase the system resistance at the correct volume and the power requirement will materially increase. Be sure that the fan driver is capable of handling an increased load within safe limits.

[Future articles will take up other servicing problems.]

Air-Borne Organisms—

[Concluded from p. 233]

A bacteriological survey conducted by Drs. C. B. Coulter and F. M. Stone of the College of Physicians and Surgeons, Columbia University, reported in the October, 1937, *American Journal of Public Health*, is also of interest. Their studies extended over a period of 18 months and covered 246 telephone booths. There seemed to be little difference in the number of air-borne organisms found on the lower east side and in the un-air conditioned telephone booths at Radio City. Pneumonia and streptococci germs were found in about the same number in unventilated telephone booths in both locations.

In the Radio Corporation of America building, for example, a series of seven booths gave counts below 60 per instrument with an average of 32, while in the Long Island Railroad station the figures for five instruments lay between 132 and 924, with an average of 500. "The difference between these two series may be due", the report states, "to the fact that the former telephone booths are in compartments which are supplied continuously with air freed of dust by air conditioning while in the latter, dust from the concourse may be carried readily into the booths."

W. F. Wells reports in the *Journal of Industrial Hygiene* "Bacteria in the 'nuclei' of evaporating droplets do not settle as rapidly as when they are attached to floating dust particles; this was determined from studies of the humidification of textile mills of Massachusetts made for the state department of health. Dust particles provide the principal source of air bacteria in the breaking and carding rooms of cotton mills, while bacteria in nuclear residues of humidifying water sprayed into the air predominate in weaving rooms. In the dusty rooms approximately five times as many bacteria for a given number of bacteria caught by the air centrifuge were observed to settle on a Petri dish in a given time as were found to settle in the heavily humidified rooms. This ratio increases to almost fifty when compared to B. coli in 'nuclei' experimentally added to the air by inoculation of the humidifying water. It is evident that 'nuclei' become widely distributed by air motion and that Petri dish exposures are useless in their determination.

"Wide distribution of 'nuclei' in buildings was further demonstrated by inoculating the humidifying water of a one room air conditioner with a culture of B. coli in the basement of the Harvard School of Public Health. Numbers of this test organism were recovered from the ends of every corridor up to the top floor of this non-mechanically ventilated three story building, in concentrations approximately 1 per cent of that in the air conditioned room.

"Experiments in an occupied standard air conditioned room have demonstrated that large numbers of typical nasopharyngeal organisms (streptococcus viridans and micrococcus catarrhalis) are projected into the air by sneezing and can be recovered in appreciable numbers from the air in the room 10 minutes after it has been vacated. An average of more than 20,000 of these organisms were discharged into the air per sneeze."

[A second article will describe various applications of air filtration to control air-borne organisms in different industries.]

Modern Piping Essential Service in the Molded Plastics Industry

By C. E. Herington*

THE molded plastics industry, conceived in the chemical laboratory, and developed to a major, far reaching industry by business initiative, presents a variety of problems to the piping engineer. Modern plastic production demands wide variation in steam and water pressures and temperatures, as well as the use of compressed air. As a result, a molded plastics plant contains interesting examples of power piping methods and equipment.

The Reynolds Spring Co. of Jackson, Mich., recently completed erection of a new plant in Cambridge, Ohio, operated by the Reynolds Molded Plastics Division of the Company, which is devoted to the manufacture of a large variety of plastic products.

At present, 20 hydraulic molding presses are installed, but the entire plant and piping layout allows for future expansion to the extent of an additional 40 presses. The presses are served by a 3000 lb hydraulic line, a 500 lb hydraulic line, a 125 lb compressed air line, a 150 lb steam line, and a 65 lb cold water line. Returns consist of a hydraulic return line, and a combined condensate and warm water return line. All lines culminate at the presses in multiple control valves manually or automatically operated for the control of press operation.

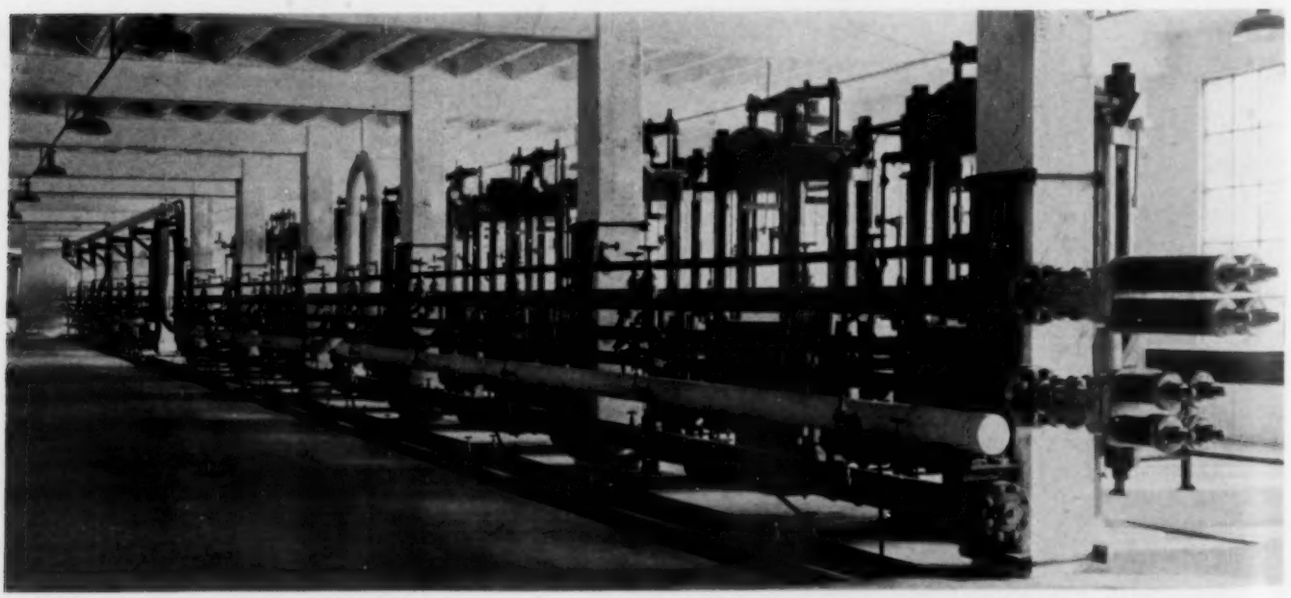
The 3000 lb hydraulic line begins at the pump bypass valve and extends to a 40 gal high pressure accumulator through a choke valve. Mains at the presses are taken off with solid forged fittings. The mains are made with long sweep bends in order to lower the friction effect and minimize the water ram. The molding presses are placed in a line parallel to the length of the building, and the hydraulic branches parallel to the line of presses are taken from the main with long sweep welded and reinforced Y fittings. At each end of both the high pressure hydraulic branch mains, forged steel spring cushions are placed to absorb any sudden shock due to press operation.

The various lines serving the presses run between the presses and are supported from concrete columns with structural supports. All hydraulic lines are rigidly anchored to prevent vibration. The individual press branch connections are made from hydraulic lines through angle connections, so as to eliminate undue shock. Outlets at the presses are divided with the necessary valves to permit isolation of the various presses.

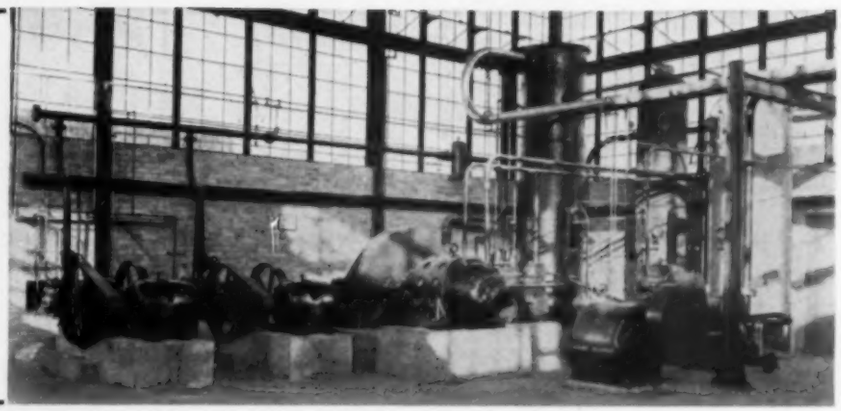
The 500 lb hydraulic line also extends from the pump to an 80 gal low pressure accumulator through a choke valve, and the main to the presses is taken off between the pump and the choke valve and extends to the presses in the same manner as the 3000 lb line. The 500 lb pressure is used to operate a press bringing the pressure against the mold to a predetermined figure, after which the 3000 lb pressure does the actual molding work. Steam is used to regulate molding temperature during fusion, cold water for cooling both molds and presses, and compressed air for removing excess material on the

*Mellott & Snyder.

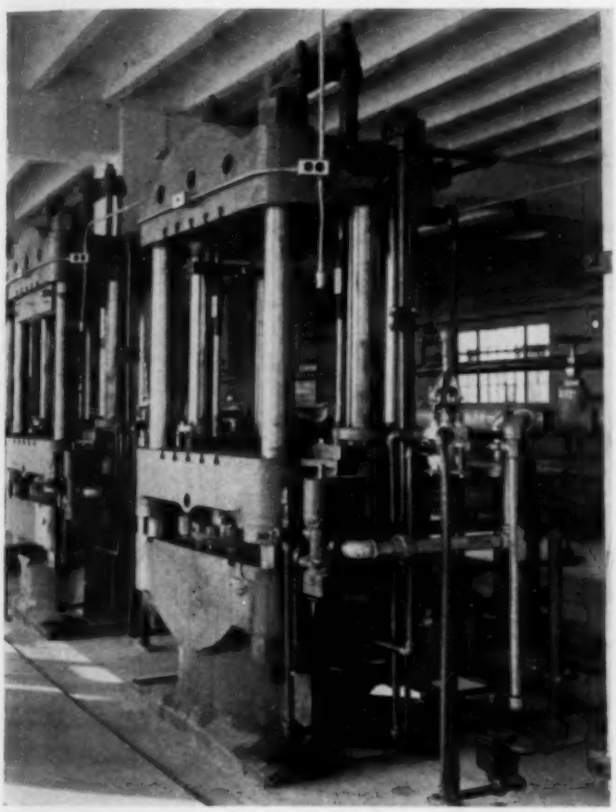
Line of 20 presses and the necessary piping, showing shock absorbers on the 3000 and 500 lb hydraulic lines. The piping has the necessary valves and take offs for an additional row of presses



Modern power piping design and layout are important features in the rapidly growing molded plastics industry. . . . The piping services in a recently completed plastics plant are described here. They include 3000 and 500 lb hydraulic lines, 125 lb compressed air, 150 lb steam, 65 lb cold water, hydraulic returns, and combined condensate and warm water return lines



Air compressors, hydraulic pumps, and accumulators which serve the presses



Close-up of one of the presses and its piping

molds after the casting has been removed. This operation method is said to give maximum efficiency in the use of pressures, steam, and water, and to reduce stress and strain on the equipment.

Both hydraulic lines return through a single line to a hydraulic return tank near the hydraulic pumps in the boiler room. Suction through the hydraulic pumps from this return tank is taken from the lower part of the tank through a line that branches to each pump. Provision has been made in this set-up for future installation of two additional 500 lb hydraulic pumps and accumulators.

Compressed air is furnished by two compressors discharging through an aftercooler into a large vertical receiver in the boiler room. The air main to the presses runs from the receiver parallel to the hydraulic lines with an additional air branch continuing to a tool and store room.

Cooling water is needed for the presses and this water is cooled in towers on the roof of the water treating room, and is then pumped through a pressure tank to the presses. Warm water leaving the presses is combined with condensate and returned through a line in a trench beneath the floor to a hot water storage tank beneath the boiler room floor. A portion of the hot water in this tank is pumped back to the cooling towers and reused. The remainder is used for boiler feed. All water used in both the boiler room and factory is treated.

The boiler is a 200 hp, 150 lb working pressure stoker fired unit. Steam is used only for factory heating and press operation. Steam passes from the boiler through a 5 in. line to an 8 in. header and is then extended to the line of presses where an 8 in. and 6 in. branch are taken off along the line. From the 6 in. branch two 4 in. branches are taken to the presses, 10 presses for each branch. The 8 in. branch was installed simply to allow for future additional presses. Expansion bends are placed in both the 8 in. and 4 in. lines. The steam condensate joins the warm water return.

Heating steam is reduced to 25 lb pressure by a reducing valve and is carried to unit heaters placed along the central columns the entire length of the building. Heating in the offices and washrooms is accomplished by means of convectors. Returns from the heating units are taken to a hot water tank under the boiler room floor by means of a return main.

The entire plant has been carefully engineered and planned throughout, contains a well equipped research laboratory, and is equipped with a new type trolley duct overhead lamp which permits sliding of the lamp supports to any point along the trolley line. Exceptional care has been taken to insure adequate light and absolute cleanliness. In all, the plant represents a thoroughly modern unit for the production of a modern product.

The piping, which is of welded construction, was engineered and installed by the Power Piping Division, Blaw-Knox Co.

An index to A. S. T. M. standards and tentative standards is available from the American Society for Testing Materials, 260 S. Broad St., Philadelphia, Pa., without charge on written request. It contains information on all of the Society's 823 standards as of January 1, 1938.

Frank Lloyd Wright Uses Floor Heating for Johnson Wax Job

WHAT is perhaps America's most interesting office building is being completed at Racine, Wis., for occupancy later this year. Designed by Frank Lloyd Wright to house executive and general offices of S. C. Johnson & Son, Inc. (Johnson's Wax), estimated total cost is around half a million dollars, of which $50,000 is for heating and air conditioning. Among its features are a floor heating system with pipe coils embedded in the slab, and complete air conditioning.

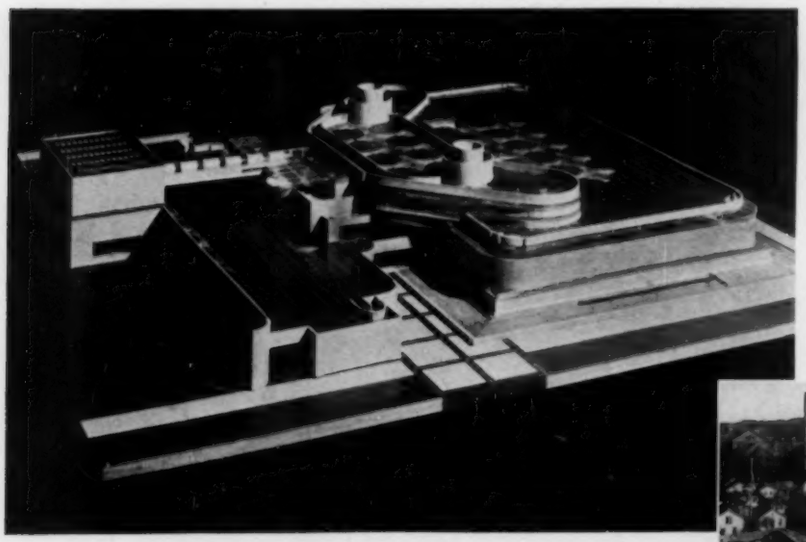
The building plan centers around one large work room for several hundred employees, with a ceiling height of 20 ft. The roof is supported by slender, hollow concrete columns with shallow, wide spreading tops, much the shape of mushrooms. Girdling this room is a mezzanine gallery on which will be offices of department heads and junior executives. Above this is a penthouse unit in the shape of three ellipsoidal links for offices of chief executives. There will be a lobby the full building height, and a theater with a capacity of 350. Utilities, rest rooms, toilets, locker rooms, etc., will be below the work room in basement spaces.

The building is entirely windowless, but natural light to supplement artificial lighting will be provided by two bands of glass completely circling the building, plus sky-light areas. These bands of glass are made up of two "tubes" with an air space between them, giving a total of three air spaces to prevent condensation in winter, and cut down heat loss or heat gain. Because these glass bands are curved at corners of the building, they had to be hand-drawn, as each sheet has a different radius.

To quote Architect Wright the building is planned to be "a highly developed synthesis of form and idea—more highly developed than has been possible in the past because it is a building without windowed walls. It consists of a great work room breathing from above through two 'nostrils'; a building having dignified character and appropriate proportions, so complete in itself that it should be in no way inferior in harmony to the ancient cathedral."

The unusual heating system comprises floor coils, supplemented by tempered ventilation air. Entirely separate from the air conditioning system, the floor coils are under the first floor slab; the floor slab rests on 9 in. of gravel fill and the steam coils, made up of 1¼ in. genuine wrought iron pipe 4 ft on centers, are placed into the top surface of the gravel fill, after which the concrete floor slab is poured. These coils consist of U-shaped pipes shop welded, with the straight runs of pipe and connections to main and return lines field welded. The coils run diagonally across the floor, with main and return lines in tunnels.

The heating medium will be steam at sub-atmospheric pressure, vacuum return, supplied from the main boiler plant several hundred feet from the building through a



Model of building showing the two "nostrils" through which it "breathes," and around the top of which the air handling equipment will be situated. The work room area is to the right; at the left is a roofed over parking area. The wide spreading tops of the mushroom shaped columns supporting the expanse of flat roof may be seen as dark circles; the lighter surrounding areas are skylights. . . . . *Below*—Construction view taken about two months ago. Steam for heating will be supplied from a central boiler plant. Exterior walls of the building are of special brick of a soft red color

3½ in. main; condensate will be returned through a 2 in. line. Steam is reduced in pressure at the entrance to the building tunnel.

One of the problems in installation of the floor coils is the method of supporting them to assure that the proper pitch will be maintained. Concrete block supports will probably be used with the pipe wired securely to each support so that the coils will remain in the proper position. The coils will be shimmed to the proper pitch before the slab is poured.

For purposes of control, the coils are to be divided into five sections, each of which will have its own control bulb to maintain a slab temperature under 75 deg. A test set-up using a floor coil of this type indicated that if the slab temperature reached 85 deg, it was too warm for comfort.

Two Separate Air Conditioning Systems

The air conditioning equipment comprises two independent systems. The building has two circular stacks, or "nostrils", each divided into an outside intake and exhaust duct, and each of which serves one of the air conditioning assemblies. The "Freon" compressors, with a total capacity of about 100 tons, will be in the roofed over parking area and refrigerant lines will run from them through the service tunnel and up through the "nostrils" to the coils, which will be installed at the top of the air shafts with the other air handling apparatus.

The treated air will be introduced to the main work room of the building through ducts running immediately under the mezzanine. These ducts are to be built of fireproof material as an integral part of the building construction, sheet metal being used only for various transition pieces. The penthouse and mezzanine offices will be heated by booster heaters in the indirect system and no radiation will be installed in them.

Three inches of cork insulation is used throughout the exterior building walls and the roof comprises a 1½ in. insulating slab with 1½ in. of cork.

The general contractor is B. E. Wiltscheck of Racine. The Westerlin and Campbell Co., Chicago, are responsible for the design and the installation of the heating and air conditioning systems. Although the systems are separate, the architect felt that undivided responsibility for these services was essential.

How Improve Operation of Vapor Heating System?

. . . . . asks a reader. Samuel R. Lewis* answers

I SHALL appreciate any light you can throw on balancing a two pipe steam system," writes a reader of H. P. & A. C. "The thermostat shuts off the gas boiler before the more distant radiators are supplied.

"The question has been asked as to whether variable orifice vent valves will be of any use. Personally I do not think they are the answer as this is a two pipe installation having a pump to return the condensate to the boiler. I am under the impression that the better method is the use of orifice plates for each radiator.

"Another factor which may be affecting the situation is the use of both old style steam radiators with inlet and outlet in the bottom together with those having inlet and outlet at top and bottom at opposite ends. I might say that all radiators have thermostatic traps on the outlets.

"I am aware that vent valves are sometimes used on two pipe radiators and I would like to know what is the objection to using the vacuum type on such applications."

Apparently the system described is a two pipe vapor system with a condensation pump. The thermostatic trap on each radiator permits the air to escape into the vented dry return main, from which an air valve allows it to pass to the atmosphere.

Installation of additional non-return equalizing air valves on each radiator would add a considerable investment cost. A vacuum due to condensation might be developed to such an extent as to draw air back through the present traps into the radiators at each cycle of steam pressure generation, and liberation of this air through graduated vent orifices would be rather slow and might even aggravate the present difficulty in heat distribution. Orifices in the radiator supply valves would do the job, but require very skillful selection and may cause water hammer.

I have improved the operation of many vapor systems of the type described by installation of control of the following type. In this scheme there are two thermostatic bulbs, one out-of-doors, the other attached to the steam pipe feeding the most distant or most sluggish radiator. The two bulbs work together in a combined instrument indoors and control the fire in proportion to the outdoor temperature, modified by the influence of the indoor bulb which will keep the burner in operation at each cycle until steam reaches and heats the pipe to which it is attached.

A less expensive but frequently conclusive scheme is to move the burner control thermostat from the air of some upstairs room as at present, to the hot water in the boiler. The water temperature can be adjusted to give the proper vapor or steam pressure-temperature which will reach the most remote radiator at each cycle and thereafter will maintain such a pressure, using longer cycles than would be allowed by the present air-responsive thermostat. The individual room temperatures are controlled as they are probably at present, by shutting off and turning on as many radiator valves as may be necessary.

*Consulting Engineer. Member of Board of Consulting and Contributing Editors.

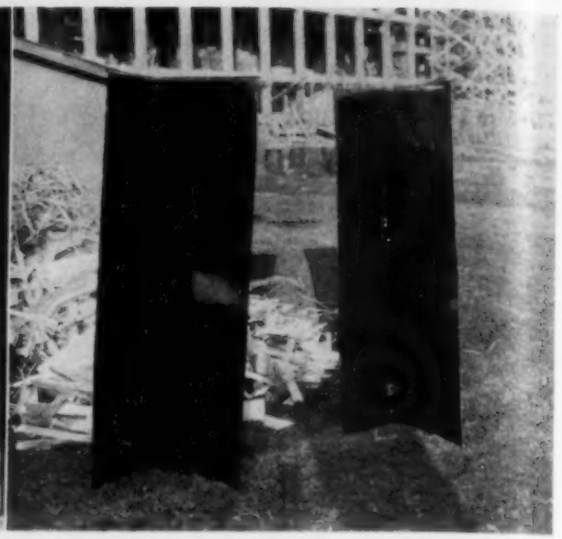


Fig. 1 (*left*)—**An advantage of the sheet metal construction is that the ducts can be made in unit sections, permitting replacements if any section wears out. Fig. 2** (*right*)—**The ducts for the Antioch general science building were protected at the time of installation with three coats of bakelite varnish followed by two coats of acidproof tar paint**

Reports Experience With Sheet Metal Fume Exhaust System

WHEN the general science building of Antioch College, Yellow Springs, Ohio, was erected in 1929, several features of its construction attracted considerable interest. Among these was the use of galvanized sheet metal for the fume exhaust system ducts. Although this was a departure from usual practice, it has met the exacting requirements and fulfilled the owners' purpose—light weight, flexibility and easy fitting.

Dr. C. S. Adams of Antioch was on the building committee for this project, and was primarily responsible for the installation. He was recently asked about the fume exhaust system duct work because another university was interested in a similar job. His following reply should be of general interest to engineers and architects:

"Some advantages of sheet metal construction for fume ducts are:

"(1) It is lighter than ceramic ducts, allowing overhead suspension in attic spaces, needs little support for risers, permitting a flexibility (additional installations) not found in the heavier ceramic installations which usually are installed as a permanent part of the building.

"(2) The ducts can be fabricated to fit the space available.

"(3) They can be made in unit sections permitting replacements if any section wears out. (See Fig. 1). We have had no replacements to date.

"(4) Sheet metal ducts permit the construction of damper systems to control the face velocity of hoods.

"(5) They are cheaper than ceramic ducts.

"Some apparent disadvantages of sheet metal ducts are:

"(1) Acid fumes, halogen fumes and other corrosive fumes will act on them. We materially eliminated this tendency by coating the ducts with three coats of bakelite varnish, followed by two coats of acidproof tar paint. This was done at the time of installation (as Fig. 2 shows) and the ducts have had no treatment since. However, I would think other methods of treatment might be known which would be more effective than the one we used. There should be some advantages in lead-lined sheet metal, for example. A few installations have been made of monel metal.

"(2) Corrosion of sheet metal ducts can be reduced to a minimum through the use of the open faced hood. This hood so dilutes the corrosive fumes with air as to render the effects negligible. A number of our ducts have been subjected to almost 24 hours service in removing the fumes from hoods in which commercial and pure research has been carried out with no failures to date. We have had some troubles recently with the aluminum blades in our damper system sticking. These were easily corrected by simply opening up the duct and oiling the blades."

Many different types of materials are at times recommended for fume exhaust system duct work. The foregoing "experience report" is therefore presented for the reader's information.

Unusual Air Conditioning Problems Met in Designing New Glass Block Building

By James N. Livermore*

A GENERAL description of The Detroit Edison Company's new Service Building was presented in the October, 1937, H. P. & A. C. In it, J. H. Walker showed the striking effect air conditioning had on the design of the structure and pointed out the reasons for many of its architectural innovations, among which are use of glass block throughout in place of windows, arrangement of office floor areas far removed from outside walls, complete absence of light courts, an air distribution construction involving the principles of a patented static pressure-displacement system, and a new type of artificial lighting with lamps set in coffers in the ceiling space. The possible structural economies with such a design were also pointed out.

The building is now under construction and while it is still impossible, from an operating standpoint, to state how well the air conditioning system was engineered, it is the purpose here to set forth some of the unusual problems encountered and their solutions.

Load Peculiarities Make Zoning Essential

Table 1 is an analysis of the cooling load. After calculating both the heating and cooling loads for morning and afternoon hours and for a variety of weather conditions, to determine required capacity of the equipment, two facts were revealed which have a decided effect on the system design. One is that in typical adjoining outer and inner bays, an area having 24 ft outside wall and 50 ft of depth, the heating or cooling load transmitted through the wall is so small as to have little or no effect on the load at the inner bay. Furthermore, with a tight outside wall having the insulating values given for glass block, internal heat sources, mainly lighting and occupants, extend the cooling period in certain parts of the building well outside the hot weather season. Corollary to these findings, it became apparent that during many hours it will be necessary to supply heat to outer bays while cooling some inner bays. Larger areas of glass block than ordinary sash are required to admit the same amount of light. Compared on this basis, the heat gain through a wall with unshaded glass block is greater than through the same wall with sash fitted with awnings, and for this reason the load due to sun becomes more important. Since the building is to be unshaded on all four sides, a change in time relation of loads can be expected in outer bays as the sun shifts from east to west. Fig. 1 shows this graphically. Note that inner bays have practically a constant load due to artificial lighting (assumed continuously in use) while outer bay loads change from hour to hour due to shift of sun and rising outdoor temperature. Thus it can be seen that there is real need for zoning or independent temperature control of many areas.

As the heating and cooling functions of the system were found to overlap through long periods, it became necessary to design on the basis of supplying the same air quantity for both heating and cooling, or for a constant air quantity to all zones at all times. In nearly every bay, the maximum cooling load, rather than heating load, determines the air quantity required, since there is a fixed low limit of entering air temperature. Thus the quantity of air to be handled is not determined by the maximum cooling load for the building as a whole, but rather by the sum of the maximum loads of all bays regardless of time. This results in a supply air demand of approximately 28,000 cfm per floor and 200,000 cfm

Table 1—Analysis of Cooling Load

	Load in Btu per Min		Total Load in Tons	Per Cent of Total Load
	Sensible	Latent		
Internal Load				
Lighting	22,717		113.59	24.44
Occupants	6,813	5,357	60.85	13.06
Glass	21,113		105.56	22.70
Walls	1,136		5.68	1.22
Roof	3,120		15.60	3.36
Infiltration	2,539		12.69	2.73
Total	57,438	5,357	313.97	67.51
External Load				
Ventilation	10,480	13,200	118.40	25.50
Fan Power	5,300		26.50	5.70
Corridor Lights	1,197		5.98	1.29
Total	16,977	13,200	150.88	32.49
Total	74,415	18,557	464.85	100.00

It has not been possible except in a few cases to take full advantage of air conditioning in evolving a building design because not many large buildings have been constructed since air conditioning became an accepted feature. This was pointed out by J. H. Walker in the October H. P. & A. C. when he described the influence of air conditioning—both architecturally and economically—on the design of The Detroit Edison Company's new Service Building. . . . Mr. Livermore's discussion completes our earlier article by describing the details of the air conditioning system, and how air conditioning was applied to a building of this type. . . . As use of glass block and totally enclosed buildings increase, the peculiar problems which appeared in the evolution of this building will present themselves many times

*Engineer, The Detroit Edison Co.

for the entire building. On an air renewal basis this averages approximately seven changes per hour.

Preliminary studies showed that it would not be possible to distribute air in such quantities from one central point on each floor without seriously impairing headroom or adding otherwise needless height to the building. For this reason three vertical supply shafts are used, each extending from the basement to the top or sixth floor. Each supply shaft was built in halves thoroughly separated by insulation, and connected at the base to two fans. One half is to be supplied with air below room temperature, the other with air above room temperature. At each floor the shaft supplies warm and cold air to a double plenum chamber compactly arranged as shown in Fig. 2.

Fig. 1 shows a typical floor divided into zones and the location of the thermostatic control point. Generally, the tops of all partitions are to stop 2 ft short of the ceiling. This makes it feasible to include in one zone all areas of like exposure without regard to partitions, the communicating space at the ceiling being sufficient to equalize the effects of any differences in interior load. There are certain exceptions, however, where partitions run full height, and in each of these cases it was considered necessary to provide a separate zone, however small the enclosed space. In some instances it was found necessary to divide areas of similar load simply to equalize the demand on the three air shafts or to keep the distributing ducts from becoming excessively large. In all, 82 zones are required.

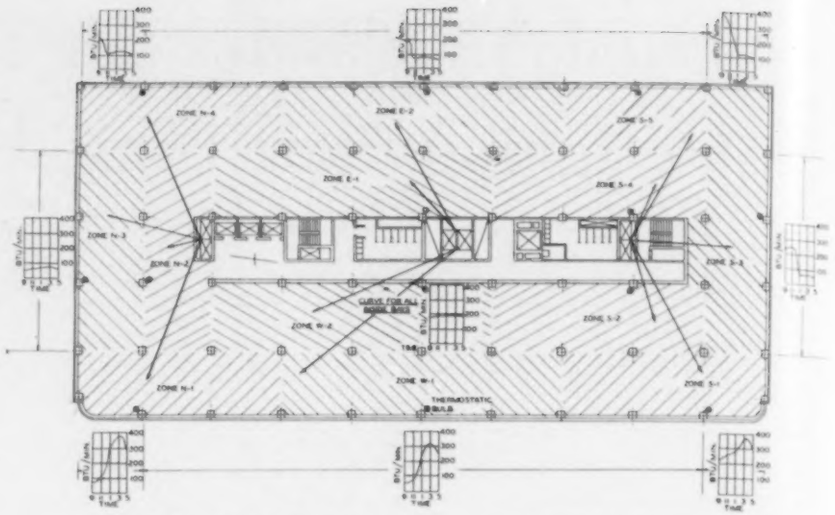


Fig. 1—Bay load curves and arrangement of zones and thermostats

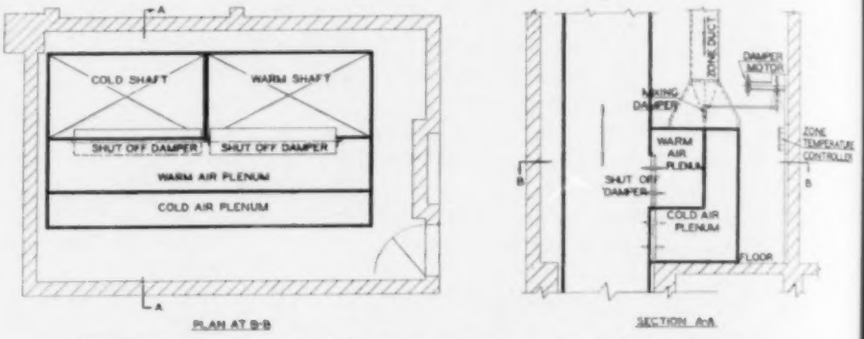


Fig. 2—Arrangement of mixing damper and supply air plenum boxes

A New Design of Mixing Damper

To obtain the temperature control desired for each zone, each one is supplied by a separate duct connecting to its own mixing damper mounted over the double supply plenum. Previous experience with this method of control has shown that the orthodox butterfly or louver type of mixing damper designed for such purposes leaves much to be desired. One inherent fault lies in the variation of total air delivered through its range of movement, as shown in Fig. 3. After some experimentation with models, the swing leaf or unbalanced type of damper was found to have practically a constant flow characteristic and was therefore adopted. A further shortcoming of the butterfly type is the difficulty encountered in making the leaves shut tight in either closed position. This has been responsible for loss of control on previous installations where, for instance, the damper was closed to the hot side but overheating resulted due to leakage. In the type adopted, extreme tightness is readily achieved by using furniture steel plate for the leaf in order to obtain perfect flatness, and stops of sponge rubber at the two sides of the damper housing. The tip of the leaf is edged with a flexible rubber lip to allow the leaf to settle on the side stops while bending in continuous contact with the damper housing. Fig. 4 shows the construction of this damper in some detail. Incidentally, all adjustable volume dampers on the zone supply ducts are also of the swing type hinged at one side of the duct, for the reason that they are less likely to rattle and produce air noise.

Fig. 5 shows these mixing dampers as arranged over the double supply plenums. In the completed building each plenum with its complement of dampers will be enclosed in a small control room, with four control rooms on each floor.

Thermostat Settings Varied Automatically

Each damper will be operated by a diaphragm motor. The motor controller is a pneumatic vapor actuated type arranged with the instrument situated in the control room and the remote temperature sensitive bulb in the occupied space, connected by capillary tubing. This arrangement takes the adjustment of controls away from the occupants and leaves their manipulation entirely to the building engineer. Where a large number of people are served by one control this seems a highly desirable provision. All sensitive bulbs are situated with their bottoms 3 ft above the floor, a height sufficient to clear the backs of chairs. It is felt that the only justification for mounting a thermostat at the conventional 5 ft level is that the occupant may more readily read the thermometer

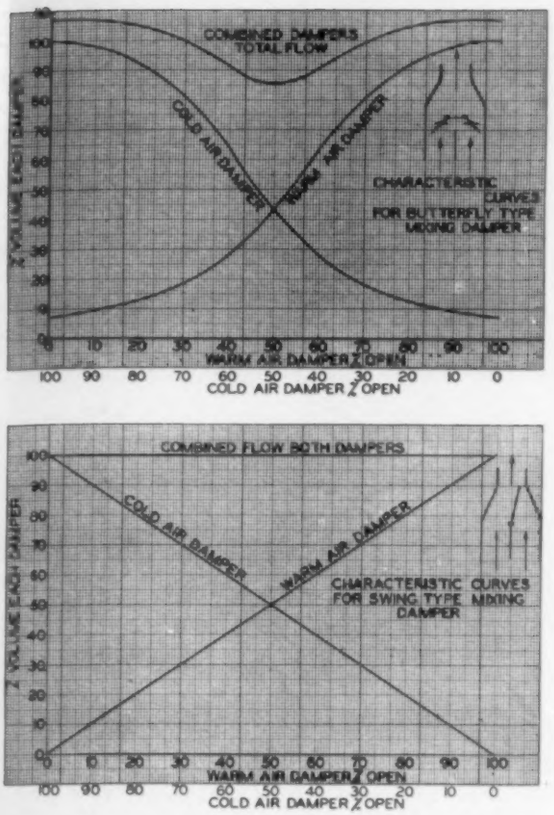

COMBINED DAMPERS TOTAL FLOW
COLD AIR DAMPER
WARM AIR DAMPER
CHARACTERISTIC CURVES FOR BUTTERFLY TYPE MIXING DAMPER
% VOLUME EACH DAMPER
WARM AIR DAMPER % OPEN
COLD AIR DAMPER % OPEN
COMBINED FLOW BOTH DAMPERS
COLD AIR DAMPER
WARM AIR DAMPER
CHARACTERISTIC CURVES FOR SWING TYPE MIXING DAMPER
% VOLUME EACH DAMPER
WARM AIR DAMPER % OPEN
COLD AIR DAMPER % OPEN

Fig. 3—Flow characteristics of mixing dampers

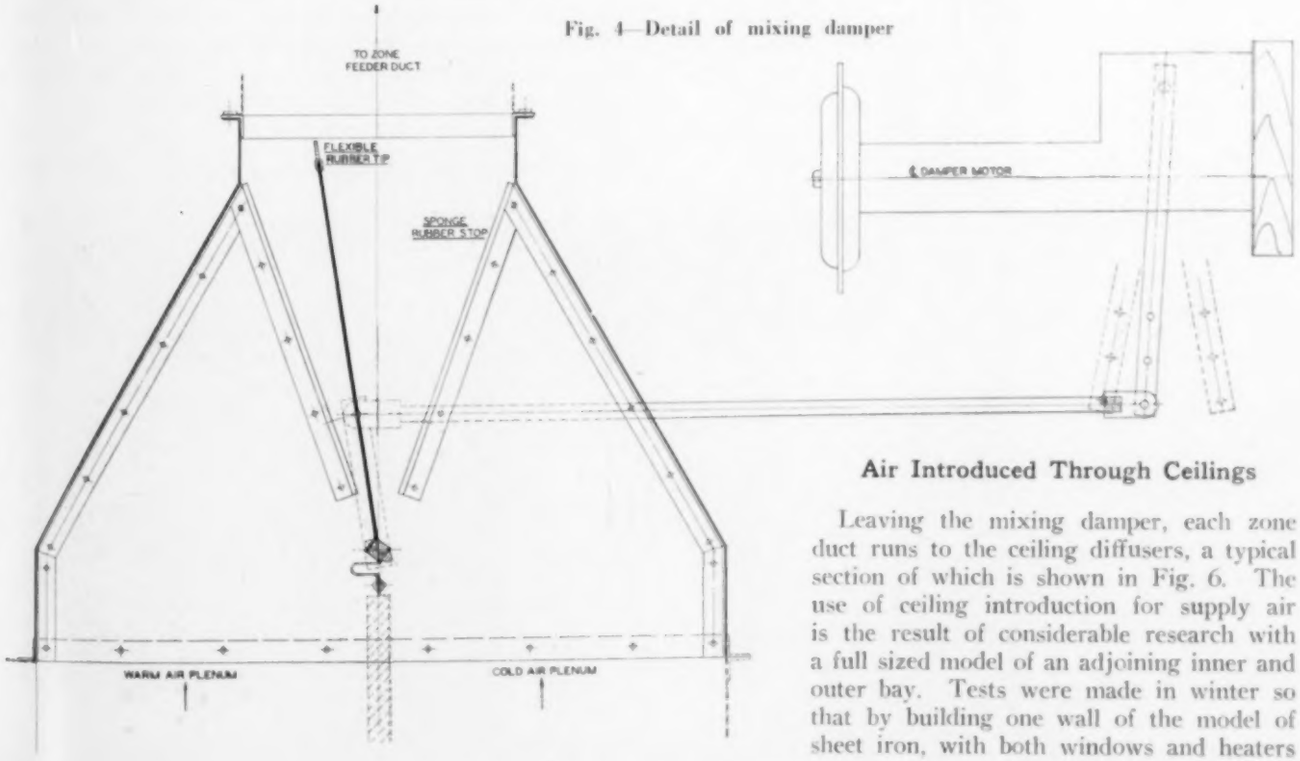
on the case, and that the control level should be at the breathing line of a seated occupant rather than for one standing. The bulbs will have no thermometers on their cases and, of course, will require no adjustment.

For the purpose of varying the inside temperature in summer with relation to the outside temperature, each of these zone control instruments has a reset mechanism which changes the temperature maintained by the controller in the occupied space. All of these mechanisms are operated pneumatically and are tied together in such manner that they can be controlled by a single master instrument on the main control board in the basement. The master controller has an outdoor sensitive bulb, and will automatically vary the settings of all the zone controllers to maintain an ideal differential between indoor and outdoor temperature throughout the hot weather season. There are maximum and minimum limits to the range over which the master controller will set the zone controller; these limits are adjustable manually, the low limit being the winter temperature adjustment.

The responsibility of attending to 82 such controls in a building where the occupant can neither juggle the thermostat nor open a window is appreciated. To simplify the task of checking up on faulty operation in any locality, multipen strip chart recorders at the main control board in the basement will show a continuous record of temperature at each of four typical points on each floor. Should a reading on any one of the records start "walking out of line," the operator should be able to anticipate many complaints.

At certain times it will be necessary to heat or cool isolated departments when the building generally is not in use. Provision for this and for certain emergencies which might arise has been made by installing a shut-off damper at each point of take-off from the supply air shafts. These dampers can be opened or closed remotely by the operator at the main control panel and will enable him to supply air to any four parts of each floor as may be required.

Fig. 4—Detail of mixing damper



Air Introduced Through Ceilings

Leaving the mixing damper, each zone duct runs to the ceiling diffusers, a typical section of which is shown in Fig. 6. The use of ceiling introduction for supply air is the result of considerable research with a full sized model of an adjoining inner and outer bay. Tests were made in winter so that by building one wall of the model of sheet iron, with both windows and heaters

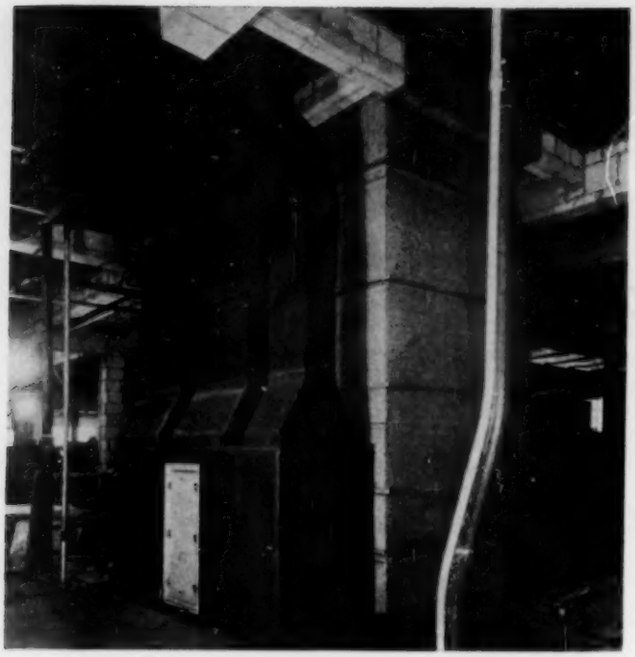


Fig. 5—Mixing dampers arranged over double supply plenums

in the space behind it, it was quite readily possible to simulate a cold or warm outside wall. It was also equipped so that warm air could be supplied to the room when it was cold, or cold air when it was warm.

All trials with horizontal projection of supply air resulted in conditions that were judged too drafty, particularly when cooling. The low ceiling height of 9 ft, 6 in., the large bays 25 ft, 0 in. square, the irregular ceiling recessed with nine 4 ft, 0 in. square lighting coffers in each bay, and the rather high rate of air supply required, all combined to interfere with its success. This was true also with efforts to introduce air upward from the window sill and a similar level at the corridor partitions. In the scheme adopted, air enters through perforated metal tile which, over the balance of the ceiling, form a part of the acoustic treatment. The velocity leaving the tile is about 40 fpm, and at head level the air becomes so diffused that its movement can scarcely be detected.

There is an aggregate length of diffusing duct of the section illustrated of over two miles in the entire building. In order to insure good distribution over this length, orifices of various sizes in the upper pressure duct were installed at approximately 12 ft intervals. Each orifice is covered by a perforated diffusing plate mounted on a threaded spindle. Raising and lowering this plate serves as a flow equalizing adjustment. A test of this design showed considerable negative flow in the porous ceiling tile until perforations were made in the diffusing discs.

This method of introduction places practically no limits on the arrangement of partitions in the future, which was considered a very important advantage.

Return Air Circuit

Leaving the occupied areas, the air may take either of two routes. In the outer bays a continuous exhaust grille at the sill or base of the glass block, supplemented by smaller openings at floor level, communicates to the ceiling space of the floor below, which in turn is connected to the suction of the return air shafts. Return air from the inner bays passes either direct to the return shafts or through louvered openings in corridor doors and thence to the same point. The return shafts draw from each floor and ceiling space and feed all the basement fan rooms.

Control of Static Pressures

As stated, a fan is required for each cold and warm half of each supply shaft. Calculations showed that the range of demand for air on either half might extend from 65 to 35 per cent of the total through both sides. To maintain a constant static pressure available at either side of each mixing damper in spite of this variation in quantity, it was necessary to provide a static pressure control. To fill this need, each fan was equipped with inlet vanes and will be driven by a three speed motor. Operating the inlet vanes will be a static pressure controller with its pressure connection located at about one-third the height of the shaft. This will so function that as the demand for warm or cold air increases on any shaft, the vanes on the warm or cold fan will be opened or closed to maintain constant static pressure. The vanes having reached the wide open or closed position, a signal light will appear on the main control board, and

Fig. 6—Section through light coffer and duct, exterior bay

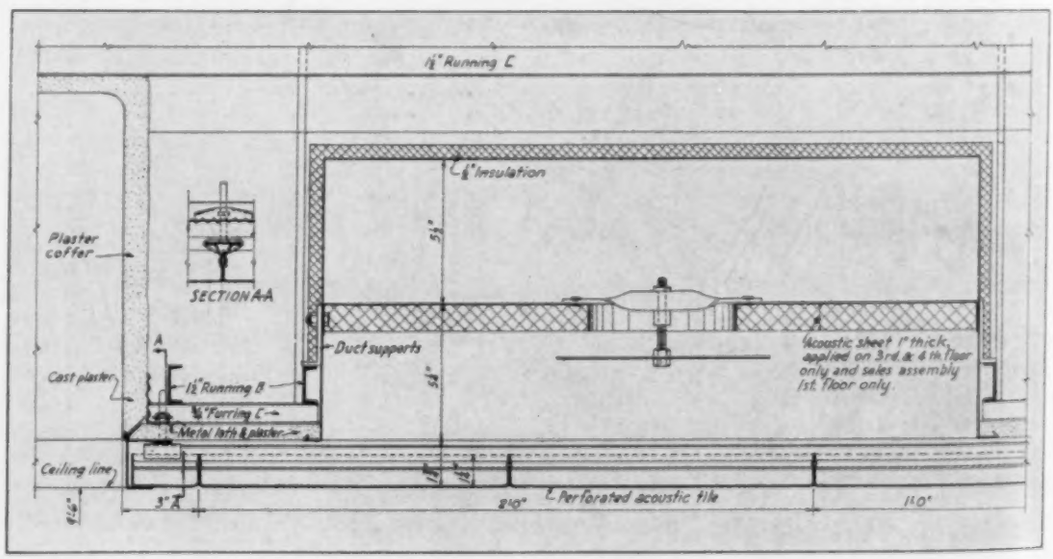


Fig. 7—Construction view of light coffers and air ducts

the operator will press the proper button to change the speed of the fan motor. The temperature of the air for each shaft is held constant by a thermostatic control. As the demand for air from any fan increases to the point where it approaches the limit of the fan's capacity, the operator sets the thermostat to a higher or lower temperature, as the case may be. This will tend to lessen the demand for air on that particular side of the shaft.

Normally, about 40,000 cfm of outdoor air or 20 cfm per occupant will be supplied purely for ventilating purposes. This same quantity will be exhausted from the building through the toilet exhaust system, the machine room, the blue printing room, and from a few shop areas. However, it would have been poor engineering indeed not to have provided a means of cooling the building with outdoor air whenever refrigeration could be dispensed with. To do this meant that outside air intakes and tempering coils had to be considerably larger than required for ventilation. Furthermore, without proper provisions, the building would become a veritable pressure chamber in which the static pressure would rise and fall with the varying demand of the cold air fans, the supply being far in excess of the constant normal exhaust. To meet this condition, two 108 in. disc exhaust fans in the pent house will be so connected to the return air shafts that their suction will communicate to all parts of the building. With these fans there will be an automatic static pressure control which, by means of dampers, will throttle their rate of exhaust sufficiently to maintain a zero differential of pressure between indoors and out.

Arrangement of Basement Equipment

Three fan rooms in the basement—one at the bottom of each shaft—make duct connections simple and lead to a clean-cut layout. Each fan has its own bank of cellulose gauze air filter units which clean both outdoor and recirculated air. Also, each fan has its own extended surface heating or cooling coil supplied by headers which carry hot and chilled water the full length of the basement. Coils and headers are so interconnected that the same water is used in them all, and as the seasons progress water temperatures are simply raised or lowered as required. Coils which were heating in winter may be cooling in summer. Provision for winter time humidification was made by installing water sprays to wet the heating coils. Water for the main heating coils will be heated in an exchanger by steam from the district heating mains, the heat of the condensate being recovered by circulating it through the domestic hot water economizer tank. Chilled water will be obtained by means of two centrifugal refrigerating machines using "Carrene No. 2" as the refrigerant, one with a rating of 360 tons when delivering 45 F water, the other 180 tons. During the early weeks of the summer, the water in the Detroit city mains is cold enough to do a good job of comfort cooling without further refrigeration. To take advantage of this the chilled water piping is so connected that city water may be used direct to the coils and thence to drain.

This outlines the essentials of an air conditioning system adapted to an unusual type of building. Being new construction, it was possible to realize many of its desirable features which, in an existing building, are usually compromised with or forgotten entirely. Among these might be named the concentration of all machinery in the basement, the incorporation of ceiling diffusers as an integral part of the ceiling, and the location of air shafts and control rooms at the most strategic points. As the use of glass block and totally enclosed buildings increase, it is felt that the peculiar air conditioning problems which appeared in the evolution of this building will present themselves many times.

Public Disappointment: Air Conditioning

The current "Fortune" turns a somewhat jaundiced editorial eye toward the air conditioning industry in an article which discusses Carrier Corp. specifically but contains general opinions on the industry. Air conditioning is termed the "prime public disappointment of the 1930's" (although Carrier's gross last year was an all time high, as were sales reported by other members of the A.C.M.A.) because the public had been led to believe it would lead us out of depression and "revolutionize indoor life." While it is no doubt true that the ultimate market for residential air conditioning is still something for the future, one is inclined to question a survey that "Fortune" reports. According to it, the answer to the question, "If you could satisfactorily air condition your whole house for $1200 or any one room for $200, would you do it?" was an emphatic "No." Without detailed information it is perhaps unwise to comment; it might be interesting to know, however, if the question were asked during a near zero wave, for instance.

"Fortune" places the blame, if blame there be, on confusion—confusion in the public mind as to what air conditioning is, confusion as to whether air conditioning should be treated as a capital goods industry or a consumer industry, confusion about marketing and distribution. It is implied that home air conditioning should be sold as a "gadget," for—says "Fortune"—there is no better way to channel the spending habits of the Amer-

[Concluded on p. 258]

Economy in Plant Heating Aided by Fuel Consumption Yardstick

By C. C. Hermann*

By determining in advance the proper temperatures to be maintained, fuel consumption for plant heating may be estimated. A yardstick is thus established and its use leads to economy in cost of heating. . . . The author tells how this procedure has been applied

TEMPERATURE control goes deeper than just having a chart showing a smooth curve for the 24 hours of the day. One of the first things I did upon taking charge of the heating of a large manufacturing plant some years ago was to set up a tentative temperature curve for the plant and then initiate supervisory control to see that the actual curve of the recorder followed the predetermined curve.

The first step was to break down the heating season into units which could be handled. The season extends from September to May in this particular plant with winter minimums down to minus 30 F. A perfect heat balance for the plant could be obtained at around 0 F so when the temperature dropped below zero it meant using live steam for heating. Naturally the below zero days were the expensive ones from the heating angle; however, even the moderate days showed a decided saving after placing the system into effect.

The degree day unit was used. The base temperature for the plant offices, pattern making departments and tool making departments was set at 65 F outside temperature. A 5 F differential was allowed between outside and inside temperature in establishing this base. For the balance of the plant, the base was set at 50 F, also referred to outside temperature. These temperatures are for the working period. During nights, Sundays and holidays the temperature was dropped to 45 degrees within so that the idle time base was established at 40 F.

The degree hours were calculated for base temperatures of 40 F for 24 hr; 50 F for 9 hr and 40 F for 15 hr; and 65 F for 9 hr and 40 F for 15 hr. Average temperatures were obtained from the watchman's hourly records over a 5 year period.

The second step consisted of determining the estimated heat loss for the various heated buildings of the plant per degree hour for the idle plant, manufacturing departments, and offices.

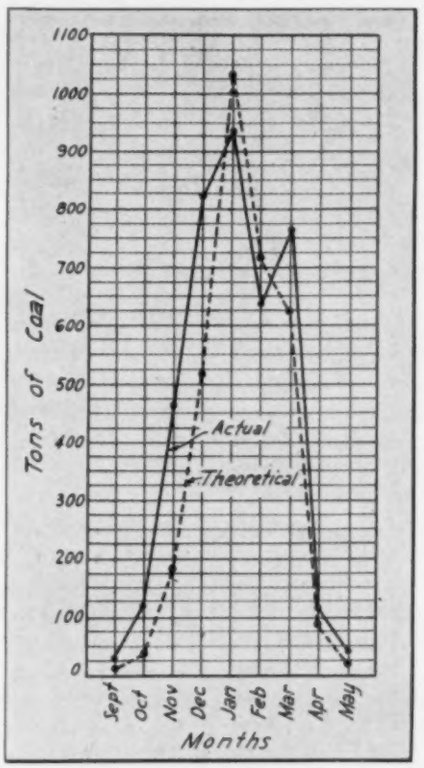


Fig. 1 — Theoretical fuel consumption compared with the actual consumption for one year

The largest variable encountered was the loss through open doors. Obviously considerable heat is lost due to trucking between buildings and wherever possible vestibules were erected to cut this to the minimum. A separate study was made to determine the possible heat loss of factory doors and this led to better design of doors, the use of electrically operated and air operated doors, as well as covered runways and vestibules. Building insulation was also shown to be of considerable importance in the heat conservation program of a manufacturing plant.

To digress for a moment, the following data relative to the heat loss due to an open door is enlightening. The doorway studied was 8 ft high and 10 ft wide with a pair of double acting doors divided in the center of the opening. The outside temperature was 28 F and the inside temperature 60 F. A moderate northwest wind was blowing and the door was open to the north exposure. The wind velocity through the door (by anemometer) averaged 390 fpm. The opening of the door to allow the average truck to pass through was taken as 30 per cent. The effective opening was therefore 24 sq ft and the infiltration was 9360 cfm. The Btu absorbed by the air per cu ft per degree rise in temperature was assumed to be 0.01957; therefore the heat loss amounted to 5861 Btu per min, equivalent to about 360 lb per hr of steam. The average time required for a truck to pass through the door and the door to close behind it was found to be 25 seconds, so that the actual cost per door per trip through it was found to be the equivalent of 2 lb of steam.

To get back to the building heat loss exclusive of the doors and ventilation, the figures were as follows: For manufacturing departments 636,200 Btu per hr per F. For offices and high temperature departments 60,100 Btu per hr per F. The total heat loss per month could then be calculated by multiplying the degree hours by these heat loss figures.

The heat loss figures could then be converted to tons of coal by dividing by 14,000,000, which provided 1000 Btu per lb of steam and 7 lb of steam per lb of coal.

*Claude B. Schneible Co. Formerly, Plant Engineer, John Deere Tractor Co.

Oxidation of Steel by High Temperature Steam

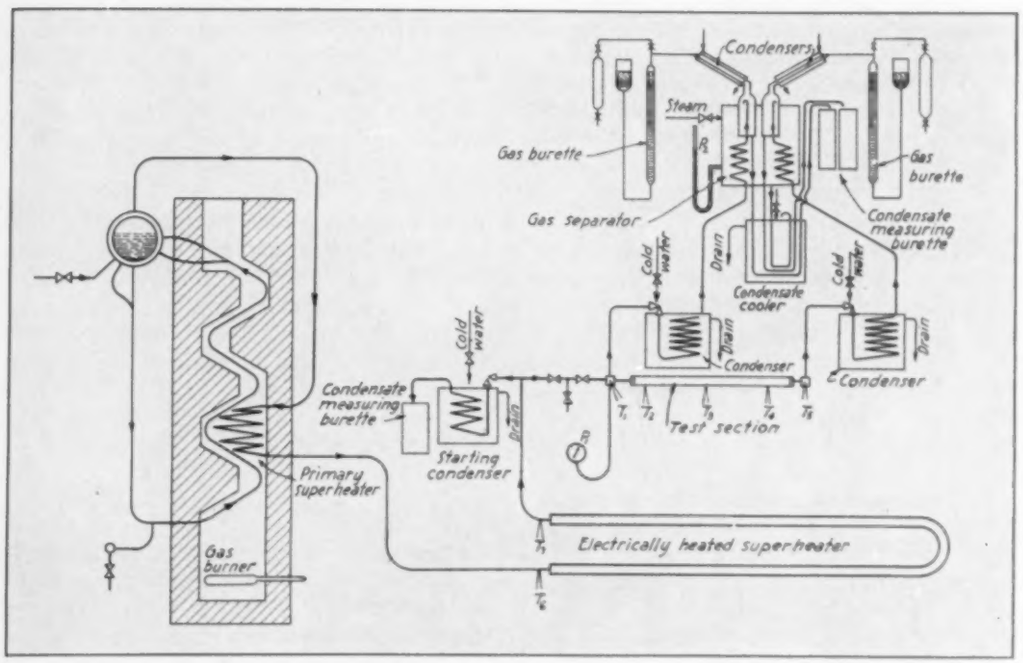


Fig. 1—Arrangement of high pressure apparatus

STEAM temperatures ranging from 1000 F to 1200 F are contemplated in power plant design because such initial temperatures would result in fuel economy and in the possibility of adopting a simple regenerative cycle without the complication of reheating. Attainment of these benefits, however, has been held up because of the difficulties encountered: (1) the strength of steel diminishes rapidly with increase of temperature; (2) creep stresses have to be provided for; (3) temperature control must be adequate; and (4) when there is a reduced rate of steam flow through superheater tubes, as during starting up or shutting down periods or low load intervals, these tubes must carry temperatures on the order of 200 F above normal. Investigation of failed superheaters has disclosed iron oxide inside the tubes, the result of dissociation of the steam when in contact with the steel tubes at about 1200 F.

A. S. M. E. Test Program at Purdue

An investigation to determine the relative resistance of the various available steels to oxidation by high temperature steam is in progress at Purdue University, and a progress report of the A.S.M.E. special research committee on critical pressure steam boilers, by Dean A. A. Potter, Prof. H. S. Solberg, and Prof. G. A. Hawkins, has been published as paper RP-59-10 of the A.S.M.E. *Transactions*. The following is a condensed account which is intended to present the most important information gleaned therefrom.

This abstract was prepared by F. E. Wertheim.

The test program includes 23 of the different compositions of steel available for high temperature steam service, and data are tabulated to show the comparative oxidation of six specimens. Plain low carbon steel was believed to be more susceptible to attack than the various alloy steels in the list, and was therefore chosen as the basic material with which to compare the behavior of the others.

The preliminary attack on the problem consisted of an attempt to obtain steam metal reaction rates on the basis of hydrogen evolution. Fig. 1 is a schematic layout of the high pressure apparatus used. The high pressure, high temperature steam was passed through electrically heated sample tubes, held at the same temperature as the steam, and uniformly heated throughout their length.

Provision was made for either condensing or sampling the steam before its entrance into the test section of tubing and for condensing it after passage through the test section. The gases in the condensate were next boiled off in a gas separator, after which the condensate flowed into a measuring burette while the gases were also collected in burettes, then stored over mercury and analyzed.

Steam for the tests was generated in a gas fired, cross drum (the drum was made out of a 10 in. locomotive axle), water tube boiler with a capacity of 50 lb per hr at 1600 lb ga and 1200 F. A primary superheater, which brought the steam to a total temperature of 800 F, at 1200 lb, is located in the boiler setting and an external, electrically heated, superheater raised steam to the final temperature required.

Although the progress report of the American Society of Mechanical Engineers' special research committee on critical pressure steam boilers on the work at Purdue University, which is abstracted here, is not strictly a matter of piping, it is closely related to the subject. Through many articles by various authors, HEATING, PIPING AND AIR CONDITIONING has presented information and data on piping for high pressure, high temperature steam; it seems desirable, therefore, to make the present discussion available to readers of H. P. & A. C. . . . . The work at Purdue is of intense interest to all concerned with high pressure and high temperature steam, and certainly to the piping engineer

Two methods of heating the independent superheater tubing were employed in these experiments. One consisted in winding coils of resistance wire, insulated with porcelain beads, around the tubing—the coils arranged in sections for either series or parallel connection with suitable control resistances. The other method was to pass alternating current through the tube itself, the tube being connected across the secondary of a transformer. Control was effected by means of a potentiometer connected to a thermocouple at the superheater outlet, operating a small relay which served to operate a circuit breaker in the primary circuit of the transformer.

All metal in contact with the steam, with exception of the test specimen, was of 18-8 stainless steel.

Occlusion Investigated

A number of initial tests showed such wide variations in the amount of hydrogen and the results, in spite of every effort, were so erratic, that more direct methods were adopted whereby the amount of oxide produced was weighed as a means of measuring the impairment of the tube.

Gases may be held in chemical combination in steel; retained on the surface of the metal; or occluded—that is, physically combined. It was thought that the erratic results in hydrogen measurement might be due to occlusion; therefore a series of experiments was undertaken to determine the rate of diffusion of gases through steel at high temperatures by means of the apparatus shown in Fig. 2. The gases were entered in the annular space between tubes and the diffusion rate was measured by collecting diffused gases from the inside chamber of the inner tube. Hydrogen at a constant pressure of 100 lb per sq in. ga passed through the wall of the inner tube (the tubes were heated to 1000 F) and was collected at atmospheric pressure, the rate of diffusion being found uniform with respect to time. The experiment was then repeated, maintaining the hydrogen at 100 lb per sq in. but varying the tube temperature within a range from 400 to 1200 F, to measure the effect of temperature upon the rate of hydrogen diffusion. The results indicated only slight diffusion below 600 F, and a rapid increase in the diffusion rate at temperatures above 600 F. Another run was made at a constant temperature of 1000 F; the gases were again collected at atmospheric pressure but the hydrogen pressure was varied from 20 to 100 lb per sq in. ga. This run showed that at constant temperature the rate of hydrogen diffusion depends directly upon the pressure differential. Similar tests were made on oxygen and nitrogen, but these gases would not diffuse through the tube at temperatures up to 1200 F. After these investigations it was concluded that the diffusion of hydrogen through the specimen was not the reason for inability to check the results of successive steam corrosion tests.

Low Pressure Steam, Superheated to 1200 F, Used for Tests

Besides the change in method already mentioned (weighing the oxide removed from the internal surface of the test specimen), a decision was made to construct test apparatus in which seven different tubes could be exposed simultaneously to identical steam conditions. Furthermore, since it had been found that steam pressure had little effect on the rate of oxidation of low carbon steel at 1100 F, later experiments were made with steam at 200 lb per sq in. ga taken from the laboratory header. The 200 lb steam was superheated to about 620 F in a gas fired superheater and then heated to a maximum temperature of 1200 F in an electrically heated coil in which alternating current passed through the tube itself, as described previously.

In preliminary test runs with low carbon tubes it was found that with temperatures as low as 800 F a layer of scale thick enough for accurate weighing was produced in 36 hr, which was accordingly adopted as a standard length of test.

Results of the Investigation

The layer of black iron oxide, Fe_3O_4, which is formed on low carbon steel is not considered to be a complete

Fig. 2—Detail of apparatus for determining the diffusion rate of gases through steel at high temperatures

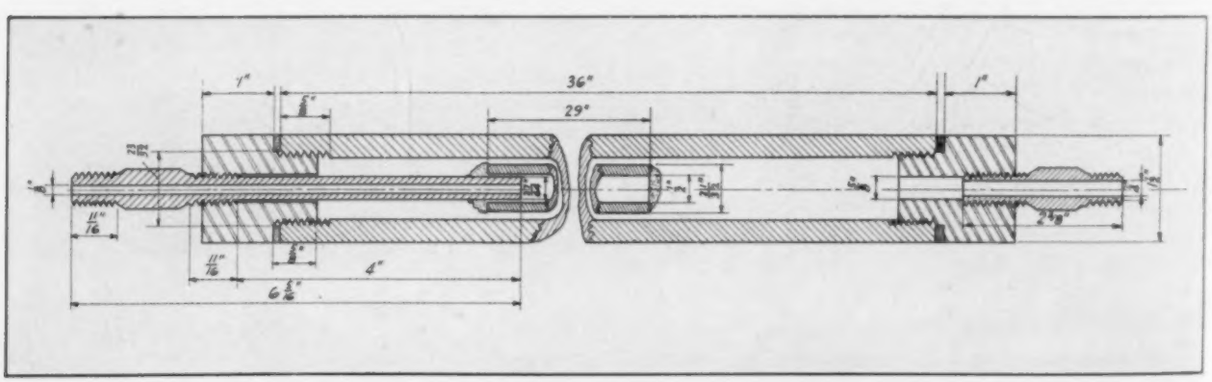


Table 1—Effect of Steam Pressure Upon the Oxidation of Low Carbon Steel by Steam

Test No.	Steam Pressure Lb per Sq In. Ga	Temperature, F		Scale Formed in 36 Hr, G per Sq In. of Internal Tube Surface
		Steam	Metal	
10	400	1100	1097	0.0962
21	397	1098	1095	0.1046
18	1200	1099	1097	0.0973
19	1200	1105	1099	0.1065

protection for the parent metal, even though the rate of reaction decreases with time. The effect of temperature on the weight of scale formed on low carbon tubes is shown by Fig. 3, on which the test data are plotted.

Table 1 indicates that at a constant temperature of 1100 F steam pressure has no effect upon the corrosion of low carbon steel.

Experiments made to determine the effect of velocity

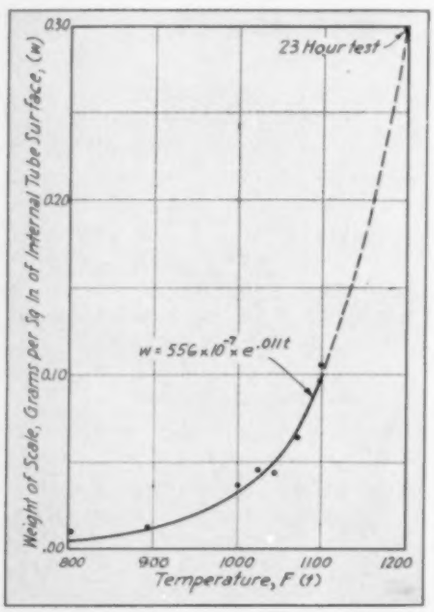


Fig. 3—Effect of temperature on the weight of scale formed (the curve is based on 36-hr tests of low carbon steel tubes which were subjected to a steam pressure of 1200 lb per sq in.)

at a constant metal surface temperature indicate that increased velocity accelerates the rate of corrosion.

The conclusions stated in the three following paragraphs are a reproduction of the authors' statements:

(1) Steam at 1200 lb and 1000 F will attack 18-8 and 25-20 stainless steels with the evolution of hydrogen and the formation of a black layer of oxide which decreases the rate of reaction as the time element increases.

(2) In view of the reaction between steam and these high chromium stainless steels, it is probable that any commercially available superheater material will be subject to a certain amount of oxidation by high pressure, high temperature steam.

(3) It is probable that for extra high steam temperatures, alloy materials should be used which will react with the steam to produce a thin, dense, tightly adherent oxide layer that offers maximum resistance to the penetration of the oxidizing agent and will not crack or spall when subjected to rapid or extreme changes in temperature.

Results of low pressure tests are given in Table 2.

These data indicate, with one exception (Tube No. 4), that the oxidation of the various alloy steels amounts to about two-thirds that of low carbon steel.

The experience of The Detroit Edison Co. is cited by reference to an experimental 18-8 stainless steel superheater which has endured 36,000 hr of service, mostly at 1100 F.

It is a fair conclusion that these investigations, which are to be continued, will prove valuable for determining the best materials to use for the 900 F to 940 F plants of the present or the 1000 F to 1200 F plants contemplated for the future.

Concrete Pipe Return Ducts

THE offices of the Clapp Baby Food Co. at Rochester, N. Y., although originally designed without provisions for air conditioning, are now equipped with an air conditioning system.

A feature of this project is the use of 396 ft of precast concrete pipe for the underground return ducts, the Portland Cement Association reports. Standard strength plain concrete pipe of the type commonly used in sewer construction was used in the branches leading to the vent risers and for all portions of the main duct smaller than 24 in. diameter. Larger sizes comprise standard strength reinforced concrete sewer pipe. Although relatively thin (the shell thickness ranges from 3 to 3¼ in.) this pipe when properly bedded is capable of withstanding external loads up to 6000 lb per lineal foot without injury.

The pipe was used in stock lengths of from 2½ to 4 ft which weigh from 28 lb per ft for the 8 in. to 453 lb. for the 33 in. All the pipe was laid by hand labor, no difficulty being encountered in either the handling or the laying.

Joints in the line were filled with portland cement mortar and when finished produced a smooth, watertight, vibrationless duct with a life expectancy as long as the building itself.

Cost of the concrete duct was nominal. Delivered prices of the pipe ranged from $0.20 per ft for 8 in. plain concrete pipe to $3.25 per ft for the 33 in. reinforced concrete pipe, and averaged $0.65 per running foot of line including Y's, curves and reducers.

Table 2—Comparative Oxidation of Six Different Steels in Contact with Steam at 1100 F and 175 Lb Ga

Tube No.	Analysis of Tube Material							Weight of Scale Formed			
								G per Sq In. of Internal Tube Surface		In Per Cent of Scale Formed in Low Carbon Steel	
	C	Mn	Si	Cr	Mo	P	S	Test No. 1	Test No. 2	Test No. 1	Test No. 2
3	0.10—0.20	0.30—0.60	.....	.....	.....	0.045	0.055	0.218	0.524	100	100
7	0.10—0.20	0.30—0.60	.....	.....	.....	0.045	0.055	0.205	0.520	.....	.....
1	0.09	0.15	1.43	.....	0.50	0.018	0.019	0.140	0.343	66	66
2	0.11	0.43	0.62	1.25	0.55	0.01	0.013	0.143	0.340	68	65
4	0.13	0.48	0.128	.....	0.51	0.013	0.012	0.162	0.469	77	90
5	0.15	0.48	0.45	3.12	0.8	0.011	0.016	.....	0.368	.....	70
6	0.104	0.41	0.43	1.99	0.56	0.019	0.011	0.141	0.321	67	62

How to Acquire a Reference Library

By George T. Condron*

SOME years ago I realized I was not getting the maximum value from technical information in past issues of publications that I was purchasing and reading. This was because of the difficulty of looking up any particular article, or group of articles, even though all of the past issues had been saved. Since then, I have built up a personal reference library arranged according to my needs which has been useful on numerous occasions.

After trying several schemes, the one described herein was adopted as best suited to my needs. The old issues of all publications that contained more than one article were dissassembled, the articles separated, and reassembled into related groups. The work involved when this system was initiated was quite an item because of the number of issues that had accumulated. However, the resulting ease of reference has more than repaid the initial effort. Chief features of the method are:

1. Back issues are disassembled and the articles separated from each other.

2. Where there is more than one sheet to an article the several sheets are cemented together at the left hand side. This prevents mixing up loose sheets. It also provides considerable reinforcement at the binding edge. After trying stapling and a number of types of paste and cement, the use of clear rubber cement was found the most satisfactory. It does not shrink or stretch the paper and holds very satisfactorily when properly applied.

3. All sheets are trimmed to a uniform size, both because of the improved appearance and for ease in handling and binding. The size adopted is 8¼x11 in. This width is enough in all but rare cases and has the advantage that the 8½x11 in. separating sheets used for indexing project ¼ in. This facilitates reference.

4. All sheets are punched on standard three ring binder spacing.

5. After sorting into the proper groups and sub-groups, the articles are bound into looseleaf volumes. It was originally planned to use the common three ring binder covers, but it was found that single sheets were apt to be torn out of this type. The use of straight, round binding posts is very satisfactory.

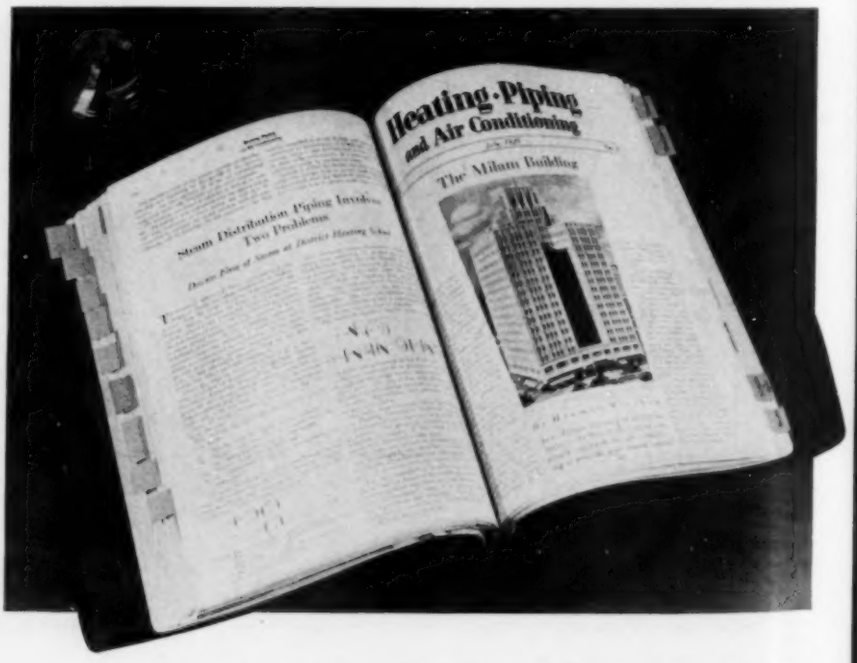
With this arrangement volumes of suitable size can be made up and the only indexing needed can be accomplished with standard separating sheets with index tabs. As the amount of accumulated material increases the size of a volume may be increased by the use of longer binding posts or more volumes may be established. In fact, the binding arrangement permits complete flexibility. The cost of binding in this manner can be suited to any pocket book for almost anything in the way of covers can be used. Stiff, hinged covers provide maximum protection to the contents, but heavy paper covers are satisfactory if cost is an important consideration.

The arrangement of the material should be suited to the user's needs. The principal classifications I employ are heat losses and gains, heating, ventilating, air conditioning, piping, refrigeration, power plants, and miscellaneous.

Advantage of Method

One advantage of the looseleaf binding is that a group of articles can be readily removed and bound together on the ring binder of a brief case, or in a common ring binder cover, if it is desirable to take them to a meeting or conference. I have found this valuable a number of times. In one case I was attending an out-of-town meeting where it seemed likely that the question of heat generated by the occupants of a room used for dining and dancing would be of importance. It was a simple matter to assemble a group of pertinent articles and reports, including one giving readings made during a University of Wisconsin "prom." These were reviewed on the train and were available for the consideration of others at the meeting. In this way it was easily demonstrated, without argument or bad feelings, that the original design had not been based on a sufficient allowance for the heat gain due to occupancy.

One of the difficulties of this method of preserving and filing articles is met where two or more articles are on the same sheet. There are a number of possible solutions which may be employed for obtaining a second copy of what is on one side of the sheet so as to have both articles complete. Frequently the publisher is willing to furnish "tear sheets" of the necessary pages.



Volumes of suitable sizes are made up, and indexed to meet the user's needs

*Mechanical Engineer.

Piping for Indirect Refrigeration

By R. C. Doremus*

This information on designing piping for indirect refrigeration systems using sodium chloride brine completes the data presented last month on calcium chloride brine and water as indirect refrigerants. . . . How to avoid corrosion and the use of corrosion retardants are also discussed

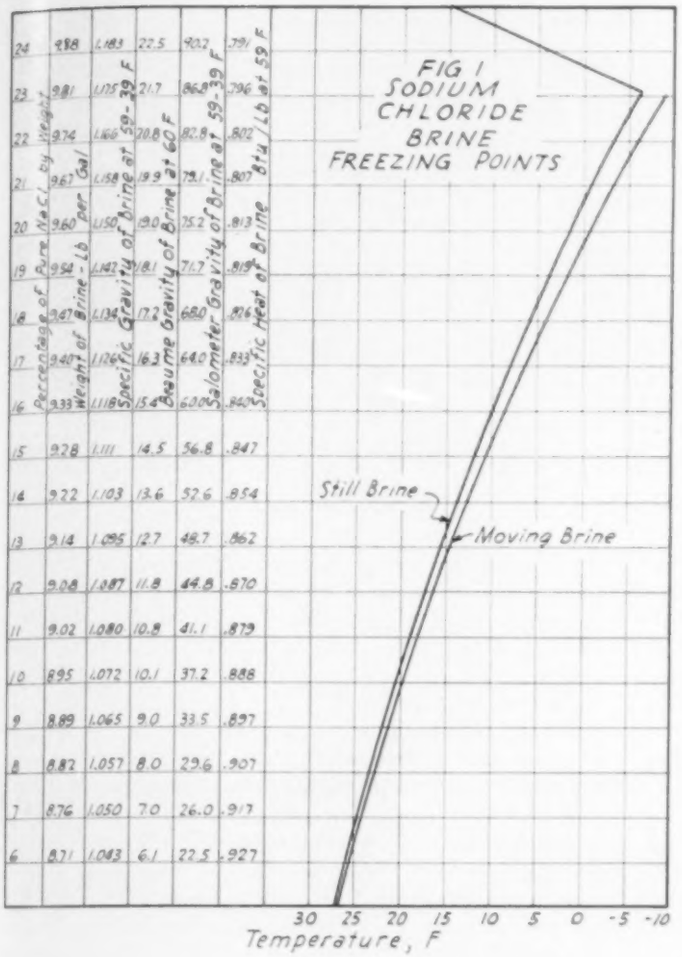


FIG 1
SODIUM CHLORIDE BRINE FREEZING POINTS
Still Brine
Moving Brine
Temperature, F
30 25 20 15 10 5 0 -5 -10
Percentage of Pure NaCl by Weight
Weight of Brine - Lb per Gal
Specific Gravity of Brine at 59-39 F
Baume Gravity of Brine at 60 F
Salometer Gravity of Brine at 59-39 F
Specific Heat of Brine
Btu / Lb at 59 F

24	988	1.183	22.5	90.2	.791
23	981	1.175	21.7	86.9	.796
22	974	1.166	20.8	82.8	.802
21	967	1.158	19.9	79.1	.807
20	960	1.150	19.0	75.2	.813
19	954	1.142	18.1	71.7	.819
18	947	1.134	17.2	68.0	.826
17	940	1.126	16.3	64.0	.833
16	933	1.118	15.4	60.0	.840
15	928	1.111	14.5	56.8	.847
14	922	1.103	13.6	52.6	.854
13	914	1.095	12.7	48.7	.862
12	908	1.087	11.8	44.8	.870
11	902	1.080	10.8	41.1	.879
10	895	1.072	10.1	37.2	.888
9	889	1.065	9.0	33.5	.897
8	882	1.057	8.0	29.6	.907
7	876	1.050	7.0	26.0	.917
6	871	1.043	6.1	22.5	.927

SODIUM chloride brine has its field of application in higher temperature refrigeration work than calcium chloride, such as from 0 to 34 F. Its strength may be increased to the minimum freezing point of approximately —6 F, but on account of the formation of slush near this point, it should not be used below 0 F.

Fig. 1 shows gravities and freezing points from which the proper strength may be determined for any application. If the brine becomes weakened by the addition of water in any process and is likely to freeze, this chart will facilitate calculation of the proper amount of sodium chloride that must be added to give the brine its correct strength and freezing point.

Like calcium chloride brine, the specific heat of sodium chloride brine varies with the strength and temperature. Average values are given on the chart to assist in making calculations, but for greater accuracy reference should be made to engineering tables of specific heats.

There is a handy rule of thumb applying to sodium chloride brine. It varies somewhat with temperature, but for a 0 F ammonia temperature, the gallons of brine circulated per minute per ton multiplied by the brine temperature range from supply to return will be a constant (25.8):

Thus, if we circulate 6 gpm per minute per ton, $t_1—t_2$ will be 4.3 F

And if we circulate 3 gpm per minute per ton, $t_1—t_2$ will be 8.6 F

And if we circulate 2 gpm per minute per ton, $t_1—t_2$ will be 12.9 F

And if we circulate 1 gpm per minute per ton, $t_1—t_2$ will be 25.8 F

Fig. 2 shows the quantity of sodium chloride brine that must be circulated for various refrigeration tonnages for certain temperature differences between brine supply and return. For an average condition, this is based on a brine supply temperature of 15 F and an ammonia temperature of 0 F, at which point the brine has a specific heat of 0.795. For other brines, definite calculations should be made for exact quantities.

For determining proper pipe sizes, pipe friction and pump head specifications, a pipe friction chart should be used. Fig. 3 shows a few curves from which a multiplier can be found. First, find what the pipe friction would be if water were being circulated, and then multiply this by the multiplier from Fig. 3 to obtain the sodium chloride brine pipe friction. As in the case of calcium chloride brine, the pipe size should be chosen to give a fluid velocity of from 4 to 7 feet per second for economical friction heads. Too small a friction head means that the pipe is too large and its cost too high. On the other hand, too high a friction head means that the pipe is too small and the saving in pipe will be more than eaten up by the increased pumping head and consequent high power bills.

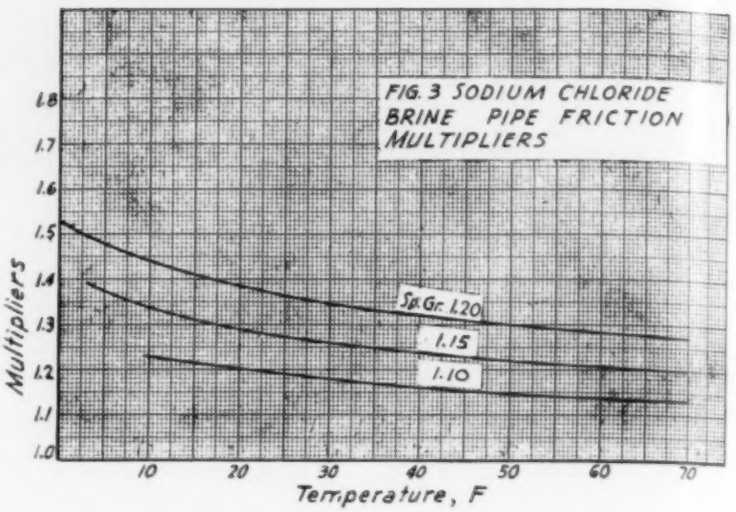
Fig. 4 is a friction head chart for water, showing friction head in feet of water.

Piping Materials

In any calcium or sodium chloride brine system, the use of dissimilar metals (such as iron-brass, iron-copper or tin-copper) should be avoided. The designer using several metals of differing electrical valence simply sets

*Chief Engineer, Detroit Ice Machine Co. Member of Board of Consulting and Contributing Editors.

up a large galvanic cell and the metal forming the positive electrode will gradually dissolve and tend to plate out on the other metal. Brine piping systems are therefore usually composed entirely of iron and steel using an all iron pump, or of brass or bronze and copper with an all bronze pump. In cases where this has been disregarded, we all know of plenty of instances where the iron, especially cast iron, was gradually dissolved, graphited and disintegrated while the brass and bronze parts stood up without any such noticeable corrosion.



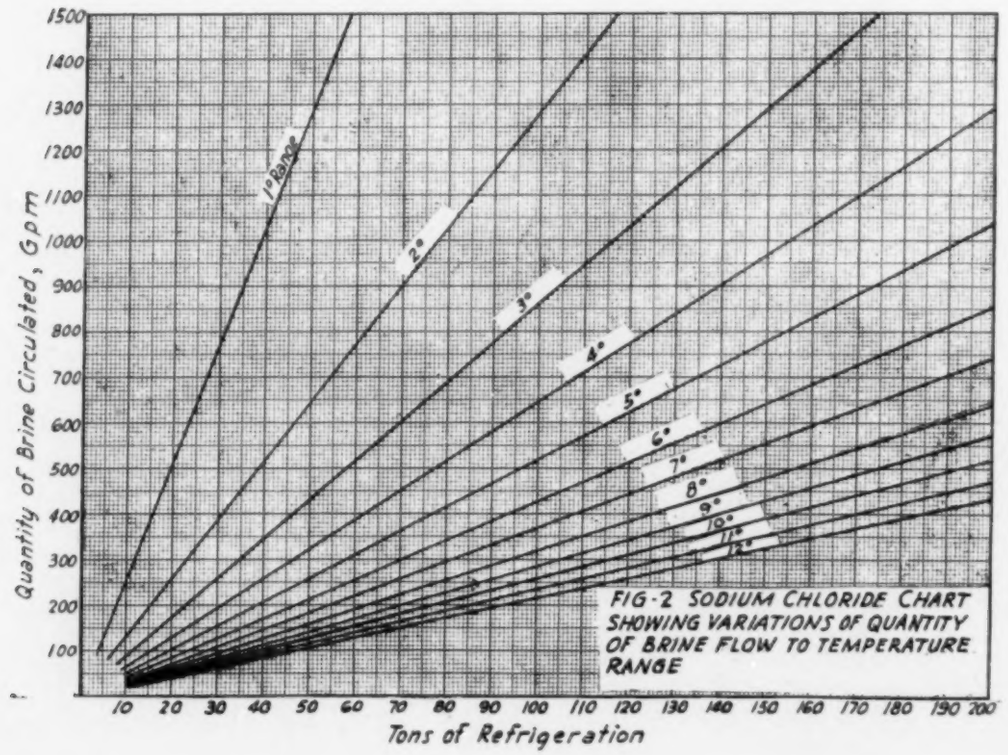
Brine Corrosion and Retardants

Even though the component parts of equipment and piping of a brine circulating system be correctly chosen, the brine is likely to be more corrosive to the metals than is water unless a corrosion retardant is used. The retardant used with calcium chloride is sodium chromate. Sodium dichromate is frequently more easily obtainable than normal chromate, but this is acid-forming so it must be normalized with alkali. Consequently, the usual dosage is 100 lb of sodium dichromate and about 31 lb of caustic soda to each 1000 cu ft of brine in the system. Tests have shown that this retardant will reduce the corrosion from 100 to 5 per cent. Most of the manufacturers of calcium chloride are able to furnish this salt either with or without chromate as desired.

Sodium chloride brine is even more corrosive to metals than calcium chloride, especially near the liquid level in an open system where the metal is alternately wetted and dried. However, corrosion may be similarly reduced by the use of 100 lb of disodium phosphate per 1000 cu ft of brine in the system. These retardants dissipate themselves about 50 per cent in a year's time and should be replenished at this rate to maintain their effectiveness.

In water circulating systems, the corrosion retardant used is sodium silicate or water glass. The dosage is 200 lb per 1000 cu ft of water in the system. This has been used with very good success for a number of years even in condenser water systems where spray ponds or cooling towers are used.

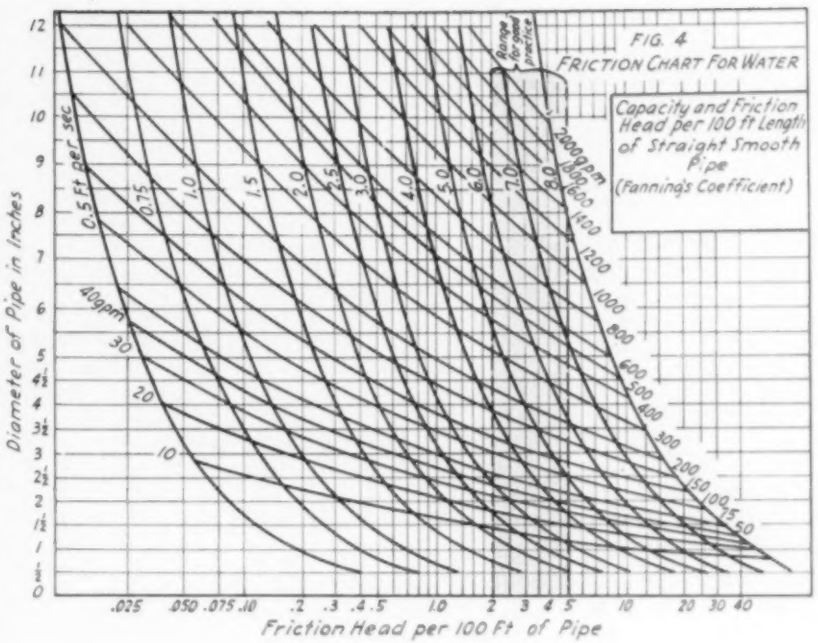
No brine of either sodium or calcium should be strongly acid or alkaline if corrosion of metals is to be avoided, regardless of whether retardants are used. An acid brine will attack any metals in the system. If the piping system is all iron and contains no brass or bronze or copper, a high alkalinity will give very good protection; yet high alkalinity is disastrous to brass and copper and their alloys. Therefore, in general, a low alkalinity is the most desirable for a low corrosion rate for all metals. On the pH scale of hydrogen ionization where 7.0 is neutral, the brine should be maintained at a pH range of 7.5 to 8.0 for brass and copper and their alloys and may be higher—8.0 to 9.0—if the system is all iron or steel.



Any laboratory or ice machine company is able to make pH tests on brine samples but the plant of 30 tons or more is large enough for the engineer in charge to have his own pH test apparatus so that he may frequently test and condition his own brine without outside assistance. If the brine is in a closed piping system, it will remain fairly constant with changes in the pH value taking place slowly. If it is in an open system and tends to turn more alkaline, its alkalinity may be reduced to the proper

point by the addition of a mild acid such as carbon dioxide gas. If it tends to turn acid, its alkalinity may be increased to the proper point by the addition of small amounts of caustic soda or lime until it comes to the desired pH range. In ice making plants, cold storage warehouses and breweries, pH brine tests should be made frequently—at least once every three or four weeks to insure keeping the brine at proper strength and at the proper alkalinity on the pH scale.

When this subject of brine corrosion of piping and equipment is more fully appreciated and more thoroughly understood than it has been, brine systems will be maintained properly as recommended by corrosion authorities. The rate of corrosion of piping and equipment will be reduced to a practical minimum which will largely eliminate the corrosion disadvantage in the brine system of indirect refrigeration.



Factors Governing Heat Transfer in Kettles

IT is well known that the metal itself offers only one of the several resistances to heat flow in transferring heat from steam to the product being processed in a kettle. The most important of these other resistances to heat flow are the films of condensed steam on the steam side, and liquid on the product side.

Since the effects of these films should not be influenced appreciably by a change in the thickness of the metal, and since, as stated above, the metal itself plays a relatively small part in determining the overall rate of heat transfer, a small change in the thickness of the metal will have only a very slight effect on the overall rate of heat transfer.

By practical tests, it is possible to determine the overall rate of heat transfer of a typical kettle, and by using the known values for the thermal conductivity of the metal, it is possible to calculate the resistances to heat flow offered by the films discussed above. Using such calculated values for the film coefficients, it is possible to estimate the overall rate of heat transfer for any thickness of metal.

For example, practical tests with a nickel steam jacketted kettle having an inner jacket 0.173 of an inch thick, and operating with 80 lb per sq in. pressure steam, showed an overall rate of heat transfer of 442 Btu per sq ft of heat surface per degree Fahrenheit temperature difference per hour, in bringing water to the boiling point.

The calculated overall film coefficient in this case was about 550 Btu, and using this value for the film coefficient it can be calculated that with nickel 0.203 of an inch thick the overall rate of heat transfer would be about 427 Btu per sq ft per degree Fahrenheit per hour. This would be approximately 96.6 per cent of the value observed for the kettle having a nickel inner jacket 0.173 of an inch thick. This means that if the area of the heating surface were kept the same, the thicker kettle would require about 3.4 per cent longer time to boil the same quantity of water, or, if it were desired to keep the heating time the same, the area of the heating surface of the new kettle should be increased by 3.4 per cent.

It should be noted that these values refer to the boiling of water. With more viscous liquids where the film coefficient would be expected to be lower, the effect of the increase in thickness of the metal would be even less than calculated for water, and, in many cases, would be negligible.

Summer Air Conditioning Season in Chicago Starts with First Day of Spring

Chicago owners and operators of air conditioning systems lost no time in starting the 1938 cooling season, for March 21—the first day of spring—was the warmest on record since 1907. The air conditioning division of Commonwealth Edison Co. answered a number of telephone requests for power service for air conditioning (particularly from owners of new systems) as the regular contracts provide for starting such service on April 1.

Carbon Dioxide Refrigeration Piping

The column heading reading "Heat of Liquid at 86 F, Btu per Lb" in the table on carbon dioxide published on page 107, February, 1938, HEATING, PIPING AND AIR CONDITIONING, should read "Heat of Liquid at 87.6 F, Btu per Lb".

E. Vernon Hill's* Page

Vacuum

"WHAT is a vacuum?" I asked this question of a student in the class, a tall, fair-haired Southerner from Alabama. His answer startled me somewhat. "A place," he said, "where there ain't nothing nohow."

I attempted to argue the point. "A vacuum," I replied, "is not a place, it is a condition." He answered, "I don't see how you can have a vacuum unless you have a place to put it."

I graded him 100 on his answer; 75 because it was evident that his conception of a vacuum was clear and definite and 25 additional because of his picturesque English.

My old Ganot's "Physics" defines a vacuum as, "a space absolutely devoid of matter" and from this definition we are forced to conclude that an absolute vacuum is an abstract conception that cannot be produced by any mechanical means at our command.

In the ordinary double piston type of vacuum pump in the laboratory, the air may be exhausted from a receiver until the pressure of the contained gases is reduced to a point where it will no longer lift the valves of the pump. We may, in this way, produce a vacuum of about 29 in. of mercury. If a higher, or lower, vacuum (I never know which to call it) is required, we must have recourse to other methods.

The Sprengel or Morren form of pump, for example, may be used. This type of apparatus uses a mercury column to produce the vacuum, a Torricellian vacuum as it is called. If a glass tube, longer than a barometer, is sealed at one end and filled with mercury and inverted, the mercury descends until its weight balances the atmospheric pressure and the space at the top of the mercury column is a "Torricellian vacuum", as near a perfect vacuum as we can produce. Utilizing this principle, Morren designed an air pump in which he could produce a vacuum of 0.1 of a millimeter of mercury, or 29.916 in.

Finkner secured even better results by filling the receiver of his air pump with oxygen containing a small amount of copper filings. After the oxygen was exhausted to the lowest point, the copper filings were heated to redness combining with the trace of oxygen left in the receiver and thus producing a very high vacuum. More recently, Crooks and other research workers, with modern methods, have been able to obtain a vacuum as low as 29.916 in. of mercury, considering a perfect vacuum as being 29.92 in.

We are not, however, so much interested in the low vacuums obtained in the laboratory as we are in the development in recent years of practical commercial apparatus that can be applied successfully in air conditioning.

Dr. William Cullen, in 1755, experimented extensively with refrigeration methods. He used the vacuum process and by means of a pump, produced fairly low temperatures. This was 10 years before the invention of the steam engine by James Watt. Dr. Cullen's work, however, was never developed beyond the laboratory stage.

At the beginning of the present century, Sir Charles Parsons took out a patent for a vacuum cooling system, using a steam jet to compress the water vapor. The steam jet, therefore, is not new, but improvements in design and increased efficiency have made it possible today to produce a vacuum in a tank containing water, of 29.75 in. of mercury, causing the water to boil at a temperature of 40 F and do this efficiently and economically. The steam jet, furthermore, if we are not particularly interested in economical operation, will carry a vacuum considerably lower than this; down to a vacuum of 29.82 in. of mercury, which corresponds to a temperature of 31.85 F.

Another very important development in the production of low vacuums is the multistage compressor. Paul Bancel and his associates have produced rather startling improvements in this type of machine in recent years. With the multistage compressor, it is now possible to handle the enormous volume of vapor necessary and still maintain a vacuum that will cause water to boil at a temperature of 40 F; and if, as in the steam jet machine, we are not particularly concerned with economy of steam consumption and condenser water, the vacuum can be carried down low enough to freeze the water in the tank. It is very probable that the development and perfection of these two methods of producing a low vacuum will markedly affect the trend of air conditioning in the immediate future.

This brings us to another point of considerable interest. We usually think of a vacuum and measure it in inches of mercury. We consider atmospheric pressure at sea level as being 14.69 lb per sq in., and if we rarify the air in some container, the pressure becomes 13, 12, and so on down until a perfect vacuum, if such could be obtained, would be zero. This is the logical, understandable, and correct method of recording pressure: pounds per square inch, absolute. It is, of course, unfortunate that we have to use the term "absolute", but so custom dictates.

In measuring a vacuum however, we do not use this logical, understandable scale of pressures but turn our scale upside down, so to speak, making atmospheric pressure zero and the scale reading increasing progressively as the pressure goes down toward zero.

We are not even content with this confusing procedure, but record pressure and vacuums in pounds per square inch, in inches of mercury and in millimeters of mercury, so the poor student who takes up the study of this subject has his difficulty in understanding fundamental principles, many times multiplied by our illogical and unnecessarily complicated methods of indicating pressures below atmosphere. I suggest, therefore, that *pounds per square inch, absolute,* be made the only standard method by all engineering societies.

John Aeberly asked me what my subject would be for my page this month. I answered him, "Nothing—that is, a vacuum". His reply was trite and not unexpected: "Your usual contribution."

*Director of Research and Education, Refrigeration and Air Conditioning Institute. Member of Board of Consulting and Contributing Editors.

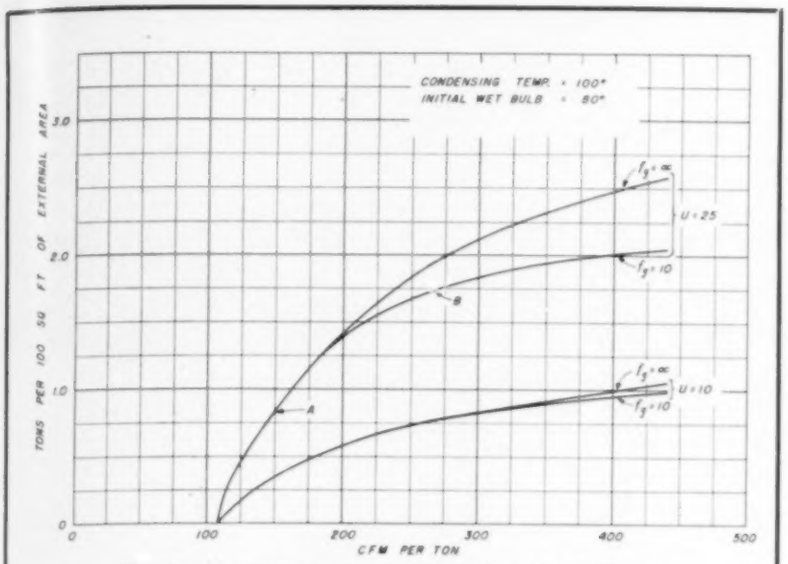


Fig. 5—Air quantity per ton plotted against tons per 100 sq ft of external area, for various coefficients of heat transfer assumed for illustrative purposes

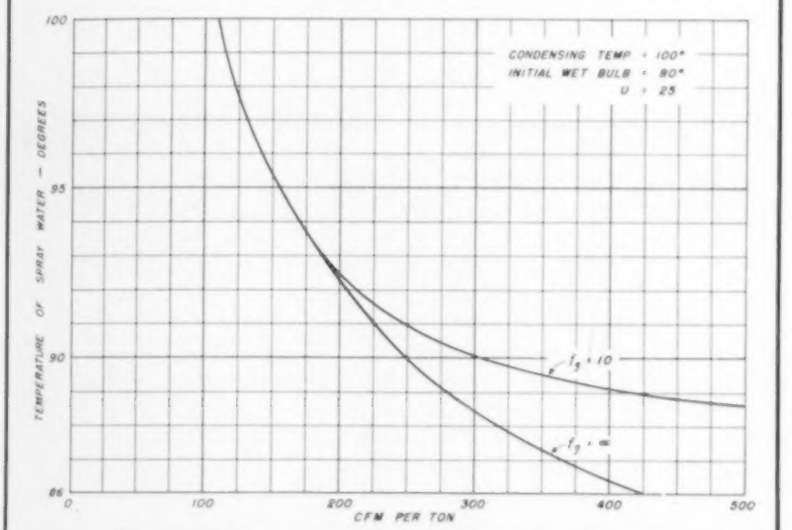


Fig. 6—The temperature of the spray water decreases as the quantity of air supplied is increased, as illustrated

The Evaporative Condenser

By William Goodman*

THE method of computing the total heat transferred in an evaporative condenser was discussed last month. Curves and tables were presented with examples showing their use. In addition, the discussion covered the rating of condensers, the selection of air quantities and coil areas.

Effect of Change in Coefficients, f_g and U, on Condenser Capacity

Generally, a large change in the value of f_g has only a small effect on the capacity of a condenser. A change in the value of U, on the other hand, has considerably more effect on the condenser performance. These statements are confirmed by Figs. 4 and 5, where groups of curves are plotted for different values of f_g and U. All of these curves are for the *same* condensing and wet bulb temperature. The ordinates of Fig. 5 are the reciprocals of those in Fig. 4. The smaller the required surface per ton, the greater will be the capacity of a condenser having a given surface area.

Notice that for each value of U, even though the value of f_g varies through as wide a range as from 10 to infinity, the change in the position of the curve is relatively small. There is, however, a considerable difference in the position of the curves for different values of U. Therefore, for all practical purposes, f_g need not be known with a high degree of precision in order to compute condenser performance with a reasonable degree of accuracy.

The value of U, on the other hand, should be known as accurately as possible because of its relatively large effect on capacity. The value of U can easily be obtained from test data by means of equation 4. The value of f_g can be computed by means of equations 1 and 2. Also, if the value of f_R, the refrigerant film coefficient, is known, the value of f_w for a particular condenser design can be calculated by means of equation 5. Resolving U into its component film coefficients enables the condenser performance to be computed for refrigerants other than the one used in the condenser test. For this purpose the film coefficients of these other refrigerants should be known with a reasonable degree of accuracy.

The curves for $f_g =$ infinity are useful for showing the limiting capacity obtainable with increasing values of f_g. The computation of the curves in Figs. 4 and 5 for f_g equal infinity is no different from the computation for finite values of f_g. From equation 2, for an infinite value of f_g, M is also infinite. Referring to Table 1, $Z = 1$. Substituting $Z = 1$ into equations 1 and 3, the balance of the computation is carried out as in example 2.

*The Trane Co. Member of Board of Consulting and Contributing Editors.

Part 2. Part 1 was published in the March, 1938, HEATING, PIPING AND AIR CONDITIONING, pp. 165-168.

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Temperature of Spray Water and Theoretical Maximum Capacity of Condensers

When computing curves like those of Figs. 4 and 5, the temperature of the spray water will be found to decrease as the quantity of air supplied is increased, and the condenser capacity increases. This is illustrated in Fig. 6, where the spray water temperatures are plotted for $f_g = 10$ and $U = 25$. These temperatures were computed as illustrated by example 2.

The temperature of the spray water falls as the quantity of air is increased, as shown in Fig. 6. At low air quantities the temperature drops rapidly for a small increase in air quantity. For larger air quantities, however, the drop in temperature is negligible even for a relatively large increase in air quantity. The maximum capacity of the condenser is obtained when the temperature of the spray water is a minimum. The minimum temperature and maximum condenser capacity can be obtained only when the quantity of air is infinite.

Although the capacity of a condenser with an infinite quantity of air has no practical significance, it is of value in indicating the limitation in capacity for an increase in air volume. It is apparent from Fig. 5 that an increase in the air volume beyond about point B has little effect on the condenser capacity.

In order to find the maximum possible capacity of a condenser as the air quantity is made infinitely large, the lowest possible spray water temperature must first be found. As shown in appendix 3, the spray water temperature is a minimum when

$$N_L = \frac{f_g}{0.24U} \quad \text{.....[6]}$$

Using this value of N in Table 2, the minimum possible spray water temperature can be found by means of Fig. 2. The theoretical maximum capacity of the condenser can then be found by substituting this minimum spray water temperature into equation 4; equation 1, should not be used as it becomes indeterminate for an infinite air quantity. The method of determining the theoretical maximum condenser capacity that the various curves of Fig. 5 approach with increasing air quantities is illustrated by example 3.

Example 3: Take the conditions of example 2 and find the lowest possible spray water temperature and the maximum possible capacity of the condenser when the air quantity becomes infinitely large.

Solution:

$$N_L = \frac{f_g}{0.24U} \quad \text{.....[6]}$$

$$= \frac{10}{0.24 \times 25}$$

$$= 1.67$$

Interpolating in Table 2, for $N = 1.67$, $\theta = 59°\text{-}5'$.

Using Fig. 2, the lowest possible spray water temperature is $t_w = 86.8$ F.

$$\frac{H}{A} = U\,(t_R - t_w)$$

$$= 25\ (100 - 86.8)$$

$$= 330 \text{ Btu per hr per sq ft of surface, maximum capacity}$$

$$\frac{330}{13{,}680} = 0.0241 \text{ tons per sq ft, maximum capacity}$$

$$\frac{1}{0.0241} = 41.5 \text{ sq ft per ton, minimum surface that theoretically can be used}$$

Example 3 also illustrates the method of finding the minimum surface area that theoretically can be used in the condenser. The actual amount of surface that should be used is, of course, greater, but a knowledge of the theoretical minimum surface is of value and interest. Notice in Fig. 4 that as the air quantity is increased, the curve approaches the value of 41.5 sq ft per ton found in the preceding example.

Also, as is evident in Fig. 4, the minimum quantity of air is required when the surface area is infinite. When the surface area is infinite, M is, according to equation 2, also infinite. For $M =$ infinity, Z is, according to Table 1, equal to one. Therefore, for $A =$ infinity, and $Z = 1$, the value of N is equal to zero according to equation 3. For $N = 0$, it is evident from Fig. 2 that $t_w = t_R$. In other words, for this condition, the temperature of the

Symbols

A = area of outside surface of condensing coil, sq ft (area in contact with spray water).
B = ratio of outside to inside surface of coil.
$\frac{A}{B}$ = area of inside surface of condensing coil, sq ft (area in contact with condensing refrigerant).
f_g = coefficient of heat transfer between air and water (air film coefficient), Btu per hr per sq ft of surface per degree Fahrenheit difference.
f_R = coefficient of heat transmission between condensing refrigerant and metal tube wall (refrigerant film coefficient), Btu per hr per sq ft per F.
f_w = coefficient of heat transmission between water and metal wall (water film coefficient), Btu per hr per sq ft per F.
G = weight of air circulated through the condenser, lb per hr.
H = total heat surrendered by the condensing refrigerant—that is, the total heat capacity of the condenser, Btu per hr.
h_1 = enthalpy (heat content) of air entering condenser, Btu per lb.
h_2 = enthalpy of air leaving condenser, Btu per lb.
h_R = enthalpy of air at a wet bulb temperature equal to the temperature of the condensing refrigerant, Btu per lb.
h_w = enthalpy of air at a wet bulb temperature equal to the temperature of the spray water, Btu per lb.
M = a quantity defined by equation 2.
N = a quantity defined by equation 3.
N_L = limiting value of N as defined by equation 6.
t_1 = initial dry bulb temperature of the air, F.
t_2 = final dry bulb temperature of air leaving condenser, F.
t_1' = initial wet bulb temperature of the air, F.
t_2' = final wet bulb temperature of air leaving condenser, F.
t_R = temperature of condensing refrigerant, F.
t_w = temperature of spray water, F.
U = coefficient of heat transmission defined by equation 5, Btu per hr per sq ft of external surface per F.
w = absolute humidity of air, lb per lb of dry air.
Z = a quantity that depends on M; see Table 1.

spray water would be equal to the temperature of the condensing refrigerant. Therefore, the theoretical minimum cfm per ton can be found by means of equation 1, by letting $h_w = h_R$, and $Z = 1$. The minimum air quantity that could theoretically be used for the conditions of Fig. 4 is 109 cfm per ton as computed in this manner. The curves of Fig. 5 converge at this value.

A knowledge of these theoretical minimum values of cfm and area is of value when designing a condenser for a given set of conditions. The actual cfm and area must, of course, be greater than these minimum values.

Final Wet and Dry Bulb Temperatures of Air Leaving Condenser

Inasmuch as all of the heat surrendered by the condensing vapor is carried away by the air, the final enthalpy of the air leaving the condenser can be found as follows:

$$H = G\ (h_2 - h_1) \qquad [7]$$

or

$$h_2 = h_1 + \frac{H}{G} \qquad [7a]$$

The final wet bulb temperature of the air corresponding to its enthalpy can be found in the psychrometric tables published in the January, 1938, HEATING, PIPING AND AIR CONDITIONING, pp. 4-6.

As shown in appendix 4, the condition of the air traveling through an evaporative condenser and in contact with the spray water at a constant temperature can be represented by a straight line on the psychrometric chart. This is illustrated in Fig. 7. Point A represents the initial condition of the air entering the condenser; point B on the saturation curve represents the constant temperature of the spray water; and point C represents the final condition of the air leaving the condenser. Point C can be found by drawing a straight line between the known points A and B. Then the final wet bulb temperature of the air leaving the condenser can be computed by means of [7a] and located on this straight line, thus locating point C. The final dry bulb temperature can then be read at C.

While air flows through an evaporative condenser, both its wet bulb and dew point temperatures increase. The dry bulb temperature, however, may either increase or decrease. If the temperature of the spray water is lower than the initial dry bulb temperature of the air, the dry bulb temperature of the air will be lowered as it flows through the condenser. On the other hand, if the temperature of the spray water is above the initial dry bulb temperature of the air, there will be an increase in the dry bulb temperature of the air as it flows through the condenser. This second case is also illustrated in Fig. 7, where point D represents a spray water temperature that is higher than the initial dry bulb temperature of the air. In the first case, the condition of the air flowing through the condenser is represented by the points on line AB, its final condition being represented by C. In the second case, the condition of the air is represented by a point on the line AD, its final condition being represented by E.

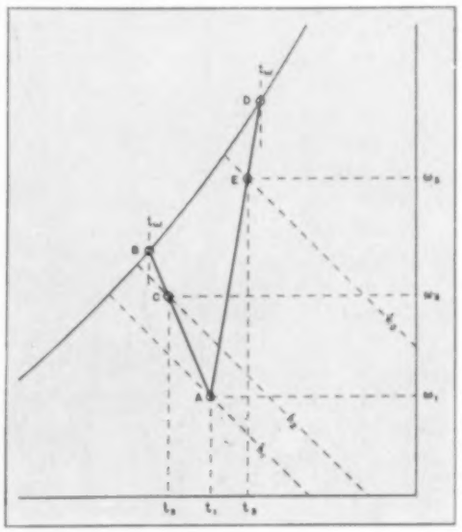

Fig. 7

The final dry bulb temperature of the air can also be computed by means of the following formula:

$$\text{Log}_e \frac{t_w - t_1}{t_w - t_2} = \frac{f_g A}{0.24G} \qquad [8]$$

Equation [8] can be used regardless of whether the temperature of the water is higher or lower than the initial dry bulb temperature of the air. This equation is obtained by integrating the following equation for the transfer of sensible heat from the water to the air.

$$0.24G\,dt = f_g\ (t_w - t)\ dA \qquad [9]$$

As shown in the following section, the change in the dry bulb temperature affects the weight of water evaporated in the condenser.

Ordinarily the weight of water evaporated in a condenser is so small that it is not important. However, in some installations—such as those in railroad cars—suffi-

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The evaporative condenser is an item of air conditioning and heat transfer equipment that has developed rapidly in recent months and is finding ever increasing usage. Very little authoritative engineering information about this type of apparatus has been generally available until now. . . . In this discussion, the author gives basic methods and equations, with examples showing their use, describes how to compute heat transferred by evaporative condensers, rating of a condenser, selection of air quantity and coil area, determination of spray water temperature and theoretical maximum capacity of condensers, calculation of final temperature of leaving air, quantity of water evaporated, etc. . . . He concludes with a few practical pointers on operation of equipment

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cient water must be carried for the duration of the run, and the weight evaporated then becomes important. Although, under these conditions, the moisture condensed from the air at the cooling coils can be drained back to the condenser, a moment's consideration will show that it must necessarily be far less in amount than the amount evaporated in the condenser.

Quantity of Water Evaporated

The amount of water that will be evaporated in a condenser is frequently computed by figuring that about 14 lb per hr of water will be evaporated per ton of refrigerating capacity. This is obtained by dividing 14,000 Btu per hr, the approximate heat surrendered by the condensing vapor per ton, by the latent heat of water, which is about 1000 Btu. However, the weight thus obtained may be in error.

The amount of water evaporated depends upon the final dry bulb temperature of the air leaving the condenser. If the dry bulb temperature is lowered, the weight of water evaporated will be larger than the approximate figure of 14 lb per hr per ton. In this case, the water that is evaporated not only carries away the heat transferred from the condensing refrigerant, but also the sensible heat lost by the air as its dry bulb temperature is lowered.

On the other hand, if the dry bulb temperature of the air is increased, the weight of water evaporated will be less than 14 lb per ton per hr. In this case, only part of the heat surrendered by the condensing refrigerant evaporates water; the balance of this heat increases the dry bulb temperature of the air. Hence, the weight of water evaporated is decreased as the dry bulb temperature of the leaving air increases.

In addition to the water evaporated, water is also carried out of the condenser by entrainment in the air stream. The loss due to entrainment depends upon the means provided for lessening this loss.

Notice in the skeleton psychrometric chart of Fig. 7 that the weight of water evaporated per pound of air decreases as the temperature of the spray water falls from *D* to *B*. But, as previously shown, the total quantity of water evaporated per ton increases as the temperature of the spray water falls. This apparent contradiction can be explained by referring to Fig. 6. For a given set of conditions, a low spray water temperature can be obtained only with an increase in the quantity of air. In other words, comparing points *D* and *B* of Fig. 7, even though the weight of water evaporated *per pound* of air decreases as the water temperature is decreased, the increase in the total quantity of air required to obtain this lower water temperature is sufficient to result in an increase in the total weight of water evaporated per ton—that is, the product of the larger air quantity and the smaller weight of water evaporated per pound of air increases.

Practical Pointers on Operation

Increasing Capacity—After a condenser has been installed, an attempt is sometimes made to increase its capacity by increasing the fan speed and the quantity of air handled. Whether or not speeding up the fan will increase the capacity of the condenser depends upon the point at which the condenser was operating originally. For example, referring to Fig. 5, if the condenser were originally operating at point *A*, an increase in the quantity of air would result in a marked increase in its capacity. On the other hand, if the condenser is already operating at point *B*, it is evident that regardless of how much the air quantity is increased, the capacity of the condenser cannot be increased much.

Effect of Noncondensable Gases—In selecting evaporative condensers, it is always necessary to allow for the effect of the ever-present noncondensable gases. In any condenser, the head pressure will invariably be higher than that corresponding to the temperature of the condensing refrigerant. Roughly speaking, the head pressure may be 5 or 10 lb higher than the pressure actually corresponding to the condensing temperature. If it is essential to prevent the condensing temperature from exceeding some previously selected value, then allowance must be made for the effect of these noncondensable gases. Therefore, in selecting a condenser, it should be selected for a head pressure that is from 5 to 10 lb lower than the maximum pressure that is permissible. For "Freon", this means selecting a condenser for a condensing temperature that is from 3 to 6 F lower than the temperature corresponding to the maximum allowable pressure.

Roof Installations—Evaporative condensers are often installed on the roof. The liquid receiver in such an installation should be protected from the direct rays of the sun. In addition to the fact that excessive pressures may be developed in the receiver because of this heat, there is also the fact that the capacity of the system will be decreased if the temperature of the liquid refrigerant supplied to the expansion valve is increased.

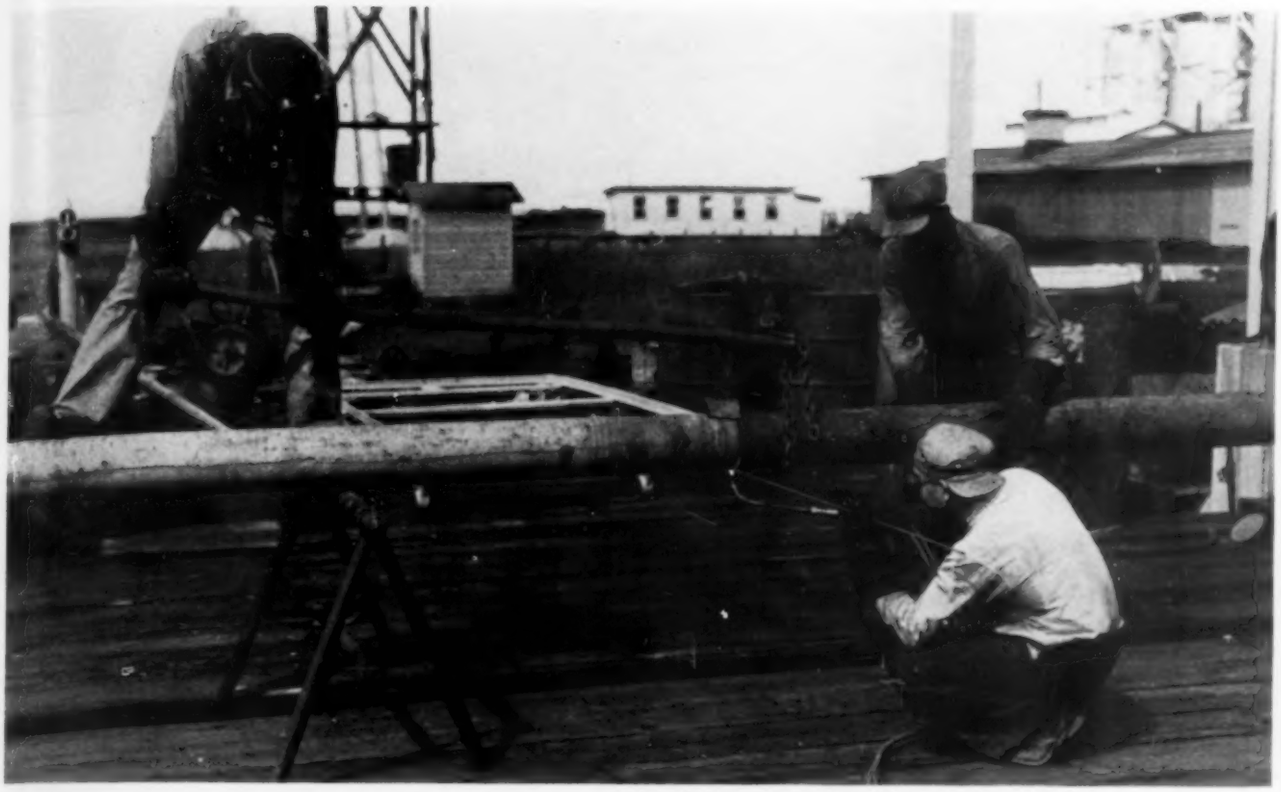
[Concluded. The appendices will be published next month.]

[Concluded from p. 245]

ican male than through "gadget appeal"; toy trains, automobile dashboards, radios, etc., are cited. To classify a service such as air conditioning with toy trains as a form of amusement seems far fetched. As to sales appeal of a gadget, it seems more accurate to say that radios, for instance, sold more widely when gadgets were removed.

According to "Fortune," a factor blocking "a big market" for air conditioning is cost, and cost is what it is because of refrigeration. Other dehumidification methods are described and it is stated that one big advantage of the moisture adsorbing process is the attitude of some electric utilities in not being too friendly toward air conditioning—for it tends to create a summer, rather than a winter, peak. This may be so in some instances; majority utility opinion is, we believe, that their business is to sell power, and if a summer peak is created, means can be found to promote winter usage to balance it—just as utilities have promoted summer air conditioning to balance the winter peak.

Despite the foregoing, the article is well worth reading for it gives those within the industry an outside viewpoint—which is almost always valuable. There are, of course, broad statements that the air conditioning engineer would be inclined to qualify; the trend in moisture adsorbing processes for instance, seems toward their use in *conjunction* with refrigeration rather than to *supplant* it.



For lining up the pipe for the tacking crew, a simple but efficient tool was developed

Welded Piping in the Sulphur Industry

By O. B. McLaughlan*

IN the Frasch method of mining sulphur, large quantities of hot and cold water, steam, compressed air, and liquid sulphur are handled. A decade ago, in plants designed for this purpose, the use of welded pipe lines and fittings was rare. In more recent plants, such as that at Grande Ecaille, La., welded piping is widely used. This plant was completed by the Freeport Sulphur Co. in 1933.

The terrain in the region of the plant required several unusual methods of construction. Being situated about 50 miles south of New Orleans in the low, swampy lands of the Mississippi Delta, transportation of men and material was entirely by means of boats and barges. A 10 mile canal connects Grande Ecaille with the slightly higher and firmer ground at Port Sulphur, which is located on the west bank of the Mississippi. Some settlement of the piling supported buildings and trestles was expected and also on account of the high pressures and temperatures of the fluids transported through the piping, welded steel construction appeared to be the preferable design.

The only source of the large quantities of fresh water required for sulphur mining was the Mississippi River. To convey this water to the plant 10 miles distant required the construction of a pipe line along the bank of the canal. This pipe line is elevated about 6 ft above the ground elevation and supported by piling trestles. From Port Sulphur to Grand Bayou, about three miles, the line consists of two 12 in. steel pipes lying parallel on the trestles. At the Bayou, an underground crossing was necessary, and this was accomplished by welding the 12 in. lines into a header from which six 8 in. lines lead downward into the mud bottom of the canal. Valves on each 8 in. line at the headers permit the shutting off of one or more of the lines for emergency repairs. On the opposite side of the Bayou, the six 8 in. lines are connected to one 14 in. line by means of another header. The single 14 in. line continues from this point to the plant at Grande Ecaille.

In the fabrication of this pipe line, three 20 ft lengths of pipe were welded together at Freeport, Texas, to form 60 ft sections which were shipped by barge to Port Sulphur. Upon arriving at Port Sulphur, the pipe was put through a machine which scraped and painted it in one pass. After being painted, two of the previously welded 60 ft sections were welded together, making a finished section 120 ft long. The 120 ft sections were then moved by barge to where they were to be used, and laid in position by a dragline machine mounted on a steel barge. When all of the pipe was in position, a tacking crew started at one end and aligned and tacked the entire line

*Master Mechanic, Freeport Sulphur Co.

The piping problems encountered in the production of sulphur are many and varied, Mr. McLaughlan told members of the International Acetylene Association at their annual convention in Birmingham last fall. He then discussed the design, fabrication, and erection of the welded piping for Freeport Sulphur's plant at Grande Ecaille, La. . . . This article, taken from his talk, shows how piping—"industry's most important transportation system"—serves in the production of sulphur

with the exception of the channel crossing. The welding operators, of whom there were three, followed the tacking crew and worked from a barge, using specially constructed platforms. The tie-in welds were made by means of position welding similar to the "bell-hole" welds on buried pipe, but all other welds on this line were made by rolling the pipe.

The backhand welding method was used on all of the welds. On the 14 in. pipe, the average time was about 20 min per weld, and about 20 min was required for moving the barge. This system of operation permitted three welding operators to complete approximately 4800 ft of pipe line per day.

In fabricating the crossings, the 8 in. pipe was welded into sections long enough to span the channels, this distance usually being about 500 ft. The individual sections were closed with "bull plugs" and tested with a hydrostatic pressure of 1000 lb per sq in. After testing, the welds were covered with forged reinforcing rings, which were welded in place with a weld along each side and a weld at each end. The entire pipe was then wrapped with tar paper and treated with a preservative. Cast iron anchor weights were bolted in place and the pipes were picked up and lashed to four barges, which towed them into position and dropped them into place. With the crossing pipes in place, the "bull plugs" were cut out, and the final tie-in welds were made. This pipe line is equipped with sliding expansion joints placed every 1000 ft and, with the exception of these joints, no leaks have occurred.

For lining up the pipe for the tacking crew, a simple but efficient tool was developed by the Freeport Sulphur Co. Among the advantages of this implement are low cost, speed of handling pipe, portability, and simplicity of construction and operation. The photograph shows this tool in position.

In the power plant, high pressure steam and air lines were fabricated from seamless tubing, the joints being ripple welded, while the low pressure lines are ordinary lap welded pipe. Most of the pipe welding in the power plant was done by the ripple method, due to the fact that experienced backhand welders were not available. The general design of the piping system is of no particular interest since the layout is mostly of a conventional nature. However, due to the necessity of uninterrupted service on the hot water system, three complete circuits were provided so that a shut-down is highly improbable.

When the piping was completed, a hydrostatic pressure of 500 lb per sq in. was imposed on the system. The boilers were then fired up and steam at 200 lb pressure was turned into the lines, after which the lines were allowed to cool and condense the steam, creating a vacuum. In spite of the complicated piping system, and the other conditions which made welding difficult, only three leaks were discovered.

For various reasons it was advisable to locate the power plant approximately one mile from the actual mining operations. Such an arrangement requires pipe lines for conveying hot water, compressed air and steam from the plant to the field, and here also the pipe lines were laid on timber trestles or bents. With the exception of the compressed air line, which carries a working pressure of 600 lb per sq in. and is made of 4 in. extra heavy galvanized pipe, all of these lines are made up of standard pipe, and are backhand welded. Because of the small diameter and the galvanized surface, the 4 in. line was ripple welded. The steam and hot water lines have a working pressure of approximately 250 lb per sq in. Those lines were welded in the conventional manner, as many joints as possible being welded by rolling, and the tie-ins being made with the pipe in its proper position.

Many Piping Problems in Producing Sulphur

The piping problems encountered in the actual production of sulphur are many and varied. It is necessary to convey the liquid sulphur from the wells to the relay station and from there to the vats directly, or to the purification plant and then to the vats. The sulphur is kept in a molten state by steam heat which is supplied by either a steam jacket or an inner line. The fabrication of these sulphur lines and the necessary fittings would be tedious and expensive without the use of welding.

On comparatively long sections of sulphur lines, a "gut line" or inner steam pipe is used to maintain the desired temperature. The "gut line" is usually 2 in. galvanized pipe joined by welding and the outer pipe which carries the sulphur is 6 in. standard. No effort is made to keep the steam pipe centered in the larger pipe, but where it enters and leaves, packing glands are supplied to prevent leakage of the sulphur, and to permit differences of expansion. Steam pressure of about 90 lb per sq in. is carried in the "gut line" to maintain the desired temperature, and the outer pipe is insulated to reduce the radiation of heat. The insulation is protected by a wrapping of tar paper which is lashed in place with wire. Any short sections, fittings, or other interruptions in the sulphur line are steam jacketed to prevent the freezing of the liquid and some of these assemblies are rather complicated, involving irregular shapes and more than one size of pipe. Welding is especially applicable to such work.

The "booster lines", as the mine water lines are commonly called, are 6 in. galvanized pipe with asbestos insulation, and a sheet iron covering, also galvanized. As the booster lines have to be moved from one location to another, they are assembled by means of standard flanges which are welded to the ends of each joint of pipe. This method of assembly permits the repeated use of the same pipe in various locations without the necessity of cutting and re-welding each time the line is moved. All of the welds on the galvanized pipe are given a coat of red lead to prevent corrosion.

Tables of Properties of Air-Vapor Mixtures Given in This Month's Heating ✦ Piping and Air Conditioning Summary Sheets

Psychrometric Properties of Air-Vapor Mixtures

By **William Goodman***

These psychrometric tables were computed from the data in the new Keenan and Keyes steam tables, and the text of the original articles from which they were summarized explained how they were computed. It is regretted that in order to reproduce them in summary sheet form it was necessary to make the figures so small, but the advantage of having the tables in this form is thought to outweigh the disadvantages.

Equations

The heat removed in cooling and dehumidifying air can be computed exactly by the following equation:

$$q = (h_1 - h_2) - (w_1 - w_2)\ h_0$$

For quick, approximate computations, the following equation can be used:

$$q = h_1 - h_2$$

In the exact equation, h_1 and h_2 are the initial and final enthalpies of air as computed by one of the equations given below. In the approximate equation, h_1 and h_2 are the enthalpies of saturated air at the wet bulb temperature of the air. These values are obtainable directly from the tables. For a discussion of the error involved, see p. 120 of original article.

The enthalpy of saturated air in column 11 of the tables was computed by means of the following equation:

$$h = 0.24t + wh_g$$

The exact enthalpy of unsaturated air can be computed by means of the tables and any one of the following formulas:

Dry bulb and dew point temperatures given:

$$h = h'' + s''\ (t - t'')$$

Also, $h = s''\ t + 1061\ w$

Wet bulb and dew point temperatures given:

$$h = h' - h'_t\ (w' - w)$$

Dry bulb and wet bulb temperatures given:

$$h = h' - \frac{s'\ (t - t')}{\frac{h_g}{h'_t} - 1}$$

Symbols

In addition to the symbols shown in the tables, the symbols used above are defined as follows. Numerical subscripts are used to indicate the point in the air conditioning cycle at which the quantity is to be taken (i.e., t_1 indicates the *initial* dry bulb temperature of the air):

- h Enthalpy of an *unsaturated* mixture, Btu per lb of dry air.
- h' Enthalpy of a *saturated* mixture at the wet bulb temperature, Btu per lb of dry air.

*The Trane Co. Member of Board of Consulting and Contributing Editors.

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Properties of Mixtures of Air and Saturated Water Vapor (Continued)

Barometric Pressure 14.696 Lb per Sq In.

All tabulated values (except temperatures and pressures) are quantities per pound of dry air in the mixture

Temp. Dry Bulb t	Pressure: Lb per Sq In. p_s	Pressure: Inches of Mercury p_s	Absolute Humidity: Pounds w	Absolute Humidity: Grains W	Volume: Dry Air Cu Ft v_0	Volume: Volume Increase for Dry Air Per Cent	Humid Specific Heat s	Enthalpy (Heat Content): Dry Air Btu h_a	Enthalpy (Heat Content): Water Vapor Btu h_v	Enthalpy (Heat Content): Air-Vapor Mixture Btu h
1	2	3	4	5	6	7	8	9	10	11
140	2.2225	4.5251	0.1113	779.3	14.86	17.8	0.2901	33.25	124.46	157.71
141	2.2826	4.6474	0.1149	804.4	14.89	18.4	0.2917	33.44	128.50	161.94
142	2.3449	4.7745	0.1185	829.7	14.91	19.0	0.2933	33.64	132.69	166.33
143	2.4093	4.9065	0.1224	856.3	14.94	19.6	0.2951	33.92	136.93	170.85
144	2.4712	5.0314	0.1263	884.1	14.96	20.2	0.2969	34.16	141.43	175.59
145	2.5370	5.1658	0.1303	912.4	14.99	20.9	0.2987	34.46	145.97	178.17
146	2.6042	5.3022	0.1345	942.1	15.02	21.5	0.3006	34.64	150.77	185.41
147	2.6726	5.4421	0.1390	972.7	15.04	22.2	0.3025	34.84	155.73	190.57
148	2.7432	5.5852	0.1435	1004.4	15.07	23.0	0.3045	35.12	160.87	195.99
149	2.8151	5.7316	0.1481	1037.0	15.09	23.7	0.3067	35.36	166.15	201.51
150	2.8886	5.8812	0.1530	1070.7	15.12	24.5	0.3088	35.60	171.62	207.22
151	2.9637	6.0341	0.1579	1105.4	15.14	25.3	0.3111	35.84	177.25	213.09
152	3.0404	6.1903	0.1632	1142.1	15.17	26.1	0.3134	36.08	183.20	219.28
153	3.1188	6.3500	0.1684	1178.9	15.20	26.9	0.3158	36.32	189.17	225.49
154	3.1990	6.5132	0.1740	1217.8	15.22	27.8	0.3183	36.56	195.47	232.03
155	3.281	6.680	0.1797	1257.9	15.24	28.7	0.3209	36.80	202.00	238.80
156	3.365	6.850	0.1858	1300.1	15.27	29.7	0.3236	37.04	208.88	245.92
157	3.450	7.024	0.1920	1343.7	15.29	30.7	0.3264	37.32	215.94	253.26
158	3.537	7.201	0.1984	1388.5	15.32	31.7	0.3293	37.52	223.23	260.75
159	3.627	7.384	0.2050	1433.9	15.34	32.8	0.3323	37.76	230.77	268.53
160	3.719	7.569	0.2119	1483.3	15.37	33.9	0.3354	38.00	238.63	276.63
161	3.813	7.759	0.2191	1533.7	15.39	35.0	0.3386	38.24	246.74	284.98
162	3.906	7.952	0.2266	1586.1	15.42	36.2	0.3420	38.48	255.34	293.82
163	4.001	8.149	0.2343	1640.1	15.44	37.4	0.3455	38.72	264.18	302.90
164	4.098	8.349	0.2424	1696.9	15.47	38.7	0.3491	38.96	273.37	312.33
165	4.201	8.557	0.2508	1755.6	15.49	40.1	0.3529	39.20	282.91	322.11
166	4.306	8.767	0.2595	1816.7	15.52	41.4	0.3568	39.44	292.91	332.35
167	4.411	8.981	0.2686	1880.1	15.54	42.9	0.3609	39.68	303.24	342.92
168	4.519	9.200	0.2781	1946.7	15.57	44.4	0.3652	39.92	314.09	354.01
169	4.629	9.424	0.2880	2016.1	15.60	46.0	0.3696	40.16	325.49	365.56
170	4.741	9.652	0.2983	2088.2	15.62	47.6	0.3742	40.40	337.16	377.56
171	4.855	9.885	0.3091	2163.8	15.65	49.3	0.3791	40.64	349.50	390.14
172	4.971	10.122	0.3203	2242.4	15.67	51.1	0.3843	40.88	362.20	403.08
173	5.090	10.364	0.3321	2324.8	15.70	53.0	0.3897	41.12	375.34	416.46
174	5.212	10.611	0.3444	2410.9	15.72	55.0	0.3955	41.36	389.79	431.15
175	5.335	10.863	0.3573	2501.0	15.75	57.0	0.4008	41.60	404.71	446.31
176	5.461	11.120	0.3708	2595.3	15.77	59.1	0.4068	41.84	419.92	461.76
177	5.590	11.382	0.3849	2694.1	15.80	61.4	0.4132	42.08	435.68	477.76
178	5.721	11.649	0.3998	2798.5	15.83	63.7	0.4199	42.32	452.13	494.45
179	5.855	11.921	0.4154	2907.9	15.85	66.2	0.4269	42.56	469.38	511.94
180	5.992	12.199	0.4319	3022.6	15.87	68.8	0.4343	42.80	487.31	530.11
181	6.131	12.482	0.4490	3142.9	15.90	71.5	0.4421	43.04	506.03	549.07
182	6.273	12.772	0.4673	3271.3	15.92	74.3	0.4503	43.28	525.69	568.97
183	6.417	13.066	0.4864	3404.7	15.95	77.2	0.4589	43.52	546.14	589.66
184	6.565	13.366	0.5067	3547.2	15.97	80.2	0.4680	43.76	567.56	611.32
185	6.715	13.671	0.5281	3696.8	16.00	83.4	0.4777	44.00	590.05	634.05
186	6.868	13.983	0.5508	3855.9	16.02	86.7	0.4879	44.24	613.58	657.82
187	7.024	14.301	0.5748	4023.8	16.05	90.1	0.4987	44.48	638.31	682.79
188	7.183	14.625	0.6004	4202.5	16.07	93.6	0.5102	44.72	663.95	708.67
189	7.345	14.955	0.6275	4392.4	16.10	97.3	0.5224	44.96	691.30	736.26
190	7.510	15.291	0.6565	4595.9	16.12	101.2	0.5354	45.20	717.13	762.33
191	7.678	15.633	0.6873	4811.1	16.15	105.4	0.5492	45.44	745.44	790.88
192	7.849	15.982	0.7201	5041.2	16.17	109.7	0.5642	45.68	775.26	820.94
193	8.024	16.337	0.7548	5285.1	16.20	114.4	0.5803	45.92	804.13	850.05
194	8.202	16.699	0.7920	5545.2	16.23	119.3	0.5974	46.16	836.40	882.56
195	8.384	17.068	0.8320	5824.9	16.25	124.5	0.6157	46.40	868.82	915.22
196	8.567	17.441	0.8750	6127.0	16.28	130.0	0.6355	46.64	902.4	949.0
197	8.753	17.821	0.9211	6449.7	16.30	135.8	0.6570	46.88	936.8	983.7
198	8.940	18.204	0.9704	6795.7	16.33	141.9	0.6803	47.12	972.8	1019.9
199	9.131	18.594	1.025	7181.7	16.35	148.6	0.7054	47.36	1010.8	1058.2
200	9.326	18.994	1.085	7597.6	16.38	155.8	0.7326	47.60	1050.5	1098.1
201	9.521	19.397	1.147	8032.9	16.40	163.4	0.7616	47.84	1091.8	1139.6
202	9.720	19.803	1.216	8523.0	16.43	171.9	0.7926	48.08	1134.9	1183.0
203	9.922	20.212	1.287	9024.6	16.45	180.9	0.8261	48.32	1180.1	1228.4
204	10.128	20.633	1.364	9561.2	16.48	190.6	0.8624	48.56	1227.1	1275.7
205	10.335	21.044	1.442	10107	16.50	200.9	0.9025	48.80	1275.1	1324.9
206	10.550	21.491	1.532	10737	16.53	212.2	0.9481	49.04	1328.1	1377.1
207	10.769	21.938	1.626	11415	16.55	224.4	0.9981	49.28	1384.1	1433.4
208	10.994	22.396	1.727	12152	16.58	237.5	1.054	49.52	1443.9	1493.4
209	11.224	22.863	1.843	12961	16.60	251.8	1.117	49.76	1507.9	1557.7
210	11.459	23.342	1.966	13846	16.63	267.3	1.189	50.00	1576.5	1626.5
211	11.700	23.832	2.102	14826	16.65	284.4	1.270	50.24	1650.9	1701.1
212	11.946	24.334	2.252	15911	16.68	303.4	1.363	50.48	1732.9	1783.4

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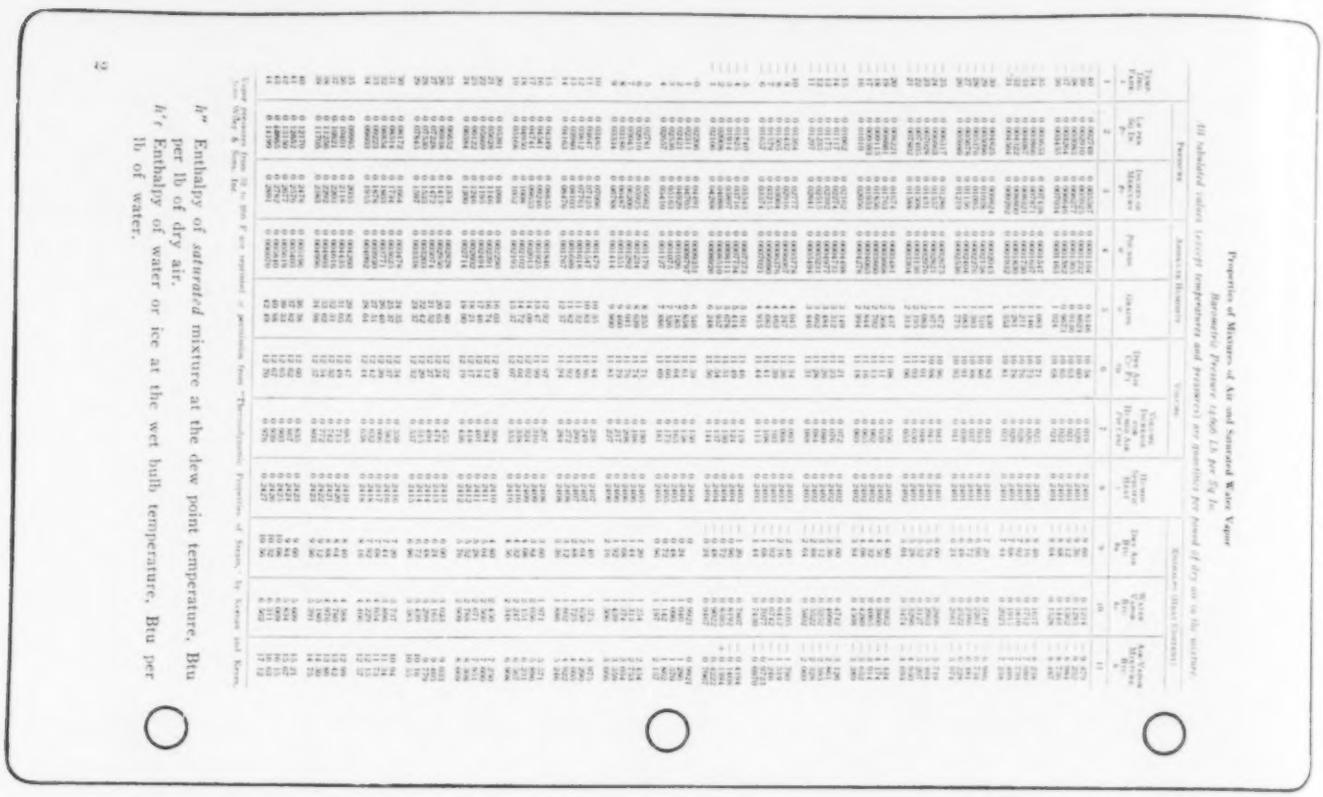
- q Heat removed from air-vapor mixture by external means, Btu per lb of dry air.
- s' Humid specific heat of a *saturated* mixture at the wet bulb temperature, Btu per lb of dry air per F.
- s'' Humid specific heat of a *saturated* mixture at the dew point temperature, Btu per lb of dry air per F.
- t' Wet bulb temperature of air-vapor mixture, F.
- t'' Dew point temperature of air-vapor mixture, F.
- w' Absolute humidity at wet bulb temperature, lb per lb of dry air.

For complete information, refer to the original article from which these data were obtained, as the purpose of these summaries is not to take the place of the article, but rather to put the essentials into convenient reference form. . . . The Editors will appreciate your suggesting what type of material you'd like presented in this "summary sheet" form. . . . Address HEATING, PIPING AND AIR CONDITIONING, 6 North Michigan Avenue, Chicago, Illinois.



Properties of Mixtures of Air and Saturated Water Vapor

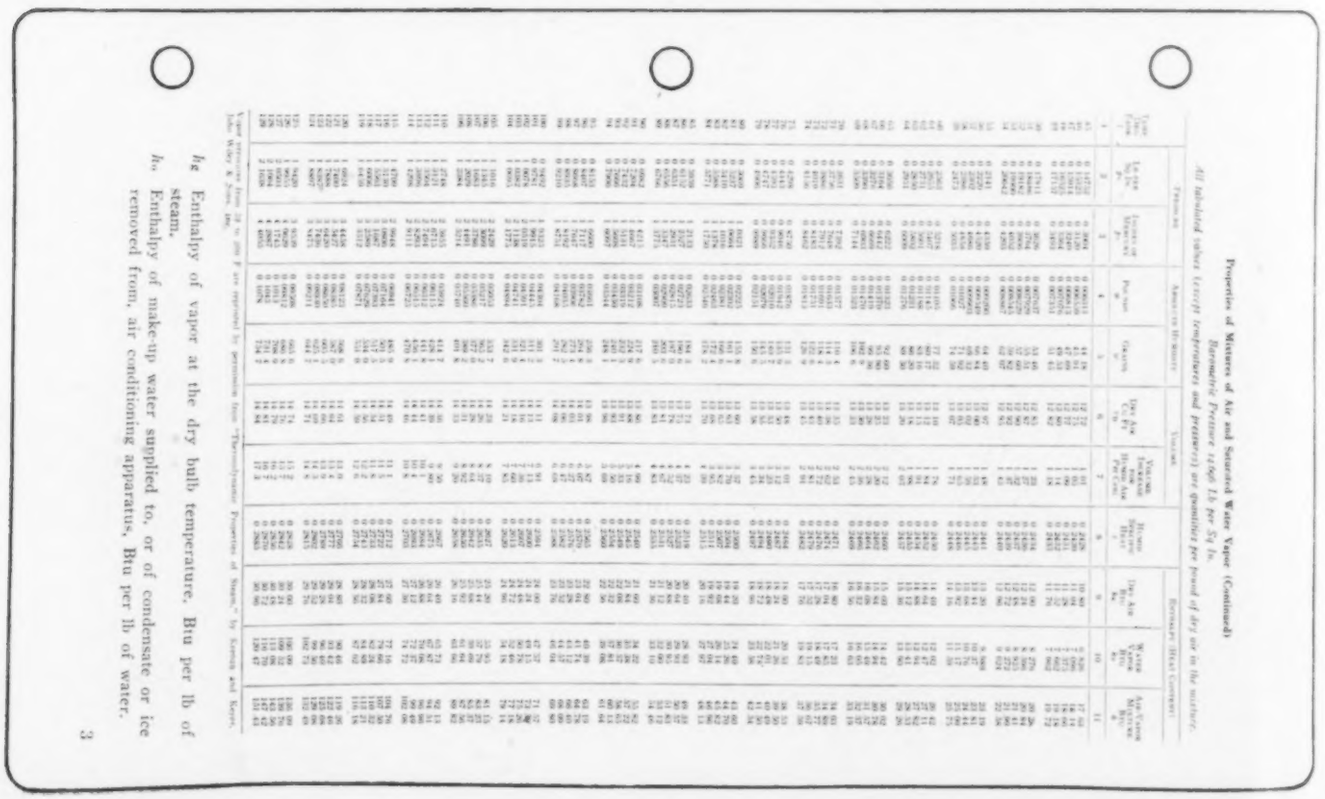
Barometric Pressure 14.696 Lb per Sq In.

All tabulated values (except temperatures and pressures) are quantities per pound of dry air in the mixture.

Tables reproduced from pp. 82 to 100 F are reprinted by permission from "Thermodynamic Properties of Steam," by Keenan and Keyes, John Wiley & Sons, Inc.

h'' Enthalpy of *saturated* mixture at the dew point temperature, Btu per lb of dry air.

h'_f Enthalpy of water or ice at the wet bulb temperature, Btu per lb of water.



Properties of Mixtures of Air and Saturated Water Vapor (Continued)

Barometric Pressure 14.696 Lb per Sq In.

All tabulated values (except temperatures and pressures) are quantities per pound of dry air in the mixture.

Tables reproduced from pp. 82 to 100 F are reprinted by permission from "Thermodynamic Properties of Steam," by Keenan and Keyes, John Wiley & Sons, Inc.

h_g Enthalpy of vapor at the dry bulb temperature, Btu per lb of steam.

h_w Enthalpy of make-up water supplied to, or of condensate or ice removed from, air conditioning apparatus, Btu per lb of water.

Booklets, Reports and Papers

How Man Regulates His Temperature

The success of a comfort air conditioning or heating system ultimately depends on how it affects the human occupant. For this reason, air conditioning engineers are more and more becoming interested in the work of the medical profession on the effect of air conditions on people and how the body loses heat and regulates its temperature.

Those who want a well rounded picture of the mechanism of heat loss and temperature regulation of the human body are referred to the Lane Medical Lectures on this subject delivered last year by Dr. Eugene F. DuBois, Medical Director of the Russell Sage Institute of Pathology, Professor of Medicine, Cornell University Medical College, and Physician-in-Chief, New York Hospital, published by the Stanford University Press, Stanford University, Cal. (paper $1.50, cloth $2.25) and comprising 95 pages and 41 text figures.

In this lecture, Dr. DuBois discussed heat production and heat loss; radiation, convection and vaporization; the significance of the surface area of the body; regulation of body temperature; and chills and fever. The author has had the opportunity to study this subject for many years and not only gives information on the work he and his associates have done, but presents much of the background of his work and the results of other researchers. Best of all, the author presents his material in a style which can be clearly understood by those unfamiliar with medical terms.

Aside from the conclusions presented, it is interesting for the engineer to learn something of the difficulties that confront the research worker in the medical field. For instance, it is a very definite problem to measure just what the surface area of the body is, and to decide whether metabolism is proportional to surface area, weight, or some other factor. Dr. DuBois concludes his chapter on significance of the surface area with the following solution, which seems quite practical: "I have often wondered what a well-trained scientist would do if confronted with the necessity of revising or re-establishing the laws of metabolism in a second Garden of Eden. The time would come when he would have to fix a general law for the level of metabolism of all the animals. He would know that protoplasm functions best at a level of about 36 to 40 C and that the animals would be exposed to environmental temperatures that averaged about 20 to 25 C. Knowing the laws of heat loss, he would probably decide that as a general working rule he would fix the basal metabolism at a level proportional to the surface area of each animal and let each species work out its own minor modifications for its own special conditions. For purposes of simplification he might make use of the fact that the surface area of different animals is roughly proportional to the two-thirds power of the weight, and let it go at that, so that the calculations could be made easily with a slide rule. This would save much time."

Air Conditioning Automobiles and Buses

L. W. Child of the Evans Products Co. spoke on winter and summer air conditioning of automobiles and buses at the recent annual meeting of the Society of Automotive Engineers in Detroit. After reviewing air conditioning principles, he pointed out that their application to an automobile or bus calls for somewhat different considerations than the problem in a building, as the space per person is considerably smaller, the air must be delivered in close proximity to the occupants (giving little chance for mixing it with surrounding air before it comes in contact with them), and the car is moving at widely varying speeds, causing widely varying amounts of outside air to leak into it and upset carefully figured conditions. The space for equipment is also a serious problem.

Mr. Child discussed in some detail problems in winter air conditioning a car and pointed out that the leakage of outside air would seem to be an uncontrollable factor. Rough tests have shown that 60 to 80 cfm delivered from outside into the body will stop leaks on most of today's four door cars at speeds up to 60 mph.

Most annoying phenomena of winter driving are formation of fog on windows and odors of wet woolen clothes and stale tobacco smoke. It was pointed out how to figure the amount of air to eliminate fogging, and it was shown that if a car contains five passengers, approximately 90 cfm is required. Heat loss calculations were also given.

Excluding leakage of air from outside into the vehicle, the cooling load inside the body will be due to the difference in inside and outside temperature, the effect of the sun, and the passengers. Each passenger will give up 260 Btu per hr. The load due to conduction through the body in summer cannot be arrived at by a simple constant times the difference between inside and outside air temperature. For winter use, a figure of 160 Btu per hr per degree difference was given as a safe figure for most five passenger cars, including the heat delivered through the dash, plus the effect of the warm air from the engine surrounding part of the body. In summer the heat from the engine is increasing, rather than decreasing this load. Also, since winter fronts are discarded and louvers opened wide, a large volume of hot air now surrounds part of the body thus increasing the average surrounding temperature. The sun generally heats the roof to a temperature higher than the surrounding air and so this part of the body load cannot be considered as a direct function of inside and outside temperature. If the roof is hotter than the surrounding air, it must be giving up heat to that air, as well as to the inside of the car. Also, as the car speed increases, the amount of heat given up to the outside air by the roof will increase, thus reducing the roof temperature and so reducing the cooling load required for this item. One test showed a drop in roof temperature from 135 F at 20 mph to 115 F at 60 mph with a corresponding 35 per cent reduction of roof load.

The floor of the car is subjected to radiant heat from the hot road, as well as hot air from the engine. This load will increase or decrease with speed depending upon the individual design of car. The remaining loads are those due to sun entering windows and warming panels, and conduction due to temperature differences, the latter generally increasing with speed.

In concluding his address, Mr. Child stated that the most logical system appears to be the use of a compressor, operated from the engine through a suitable variable speed drive. Such a system will require a minimum space and not over 2 hp, a figure well below the power required to operate the present radiator fan.

Health Aspects of Air Conditioning

Air conditioning is largely a matter of comfort and happiness and is more closely allied to health than to luxury, Dr. C. P. Yaglou, of the Harvard school of public health, declared in a talk given under the auspices of the General Electric air conditioning institute in New York City. Dr. Yaglou pointed out that air conditioning's aid in combatting fatigue and disease was not so much a direct attack as it was a strengthening of natural physical processes.

"Among the important environmental factors affecting equilibrium of health," he said, "are temperatures and temperature changes, drafts, humidity, barometric pressure, and purity of air with respect to harmful dusts, gases, odors, and disease producing organisms. No one is capable of escaping the influence of these factors.

"The importance of air conditioning varies considerably in different persons, and cannot be directly evaluated in terms of health, partly because we have no satisfactory health criteria, and partly because there are many co-existing influences unrelated to air conditioning that are difficult to control. At the

present stage of our civilization we are not content with merely avoiding discomfort, but we demand the same standards of purity in the air we breathe as in food and drinking water."

Dr. Yaglou concluded with a survey of the effectiveness of modern air conditioning in therapeutics, particularly its employment in hospitals and clinics, indicating that this should be a far reaching development.

"Cooling of operating and post operative rooms is fast becoming a regular technique," he said. "It protects the patient against heat stroke and improves his recuperating power in hot weather. It also improves the efficiency of the surgeon. Year 'round conditioning is often necessary for reducing the risk of explosion. In nurseries for institutional care of premature infants, the requirements for humidity are so rigorous that they can only be maintained by the use of air conditioning systems.

"An interesting trend in the treatment of diseases is the increasing number of pressure chambers for the study of syndromes that are believed to be initiated or accelerated by variations of atmospheric pressures. These chambers must be fully conditioned."

Ventilation of Mines

In view of the great importance of mine ventilation to both the workman and the operator, a survey of the ventilation characteristics of several typical Illinois mines has been made during the past year or two, and is described in Bulletin No. 297, "Ventilation Characteristics of Some Illinois Mines," by Cloyde M. Smith, which has just been issued by the Engineering Experiment Station of the University of Illinois. As a result of the survey a number of recommendations are made for the improvement of the ventilating systems in efficiency and economy of operation.

Until August 15, or until the supply available for free distribution is exhausted, copies of the bulletin may be obtained without charge upon application to Engineering Experiment Station, Urbana, Ill.

Book Reviews

Alignment Charts for Moist Air

["Fluchtentafeln für Feuchte Luft," by Dr. Ing. Herbert Jahnke, published by Julius Springer, Berlin. 1937. Price, RM 12.60. Reviewed by Dr. F. E. Giesecke, Director, Texas Engineering Experiment Station.]

Doctor Jahnke points out that Professor Mollier's *i-x* charts for moist air (V. D. I. Journal, 1929, Vol. 73) give data relating to temperature, moisture content, heat content, cooling limits, dew point, etc., but do not give data relating to total volume, specific volume, and specific weight of moist air. Also, that, while Professor Mollier's charts could be expanded to include the additional data, it would complicate them, and he recommends the use of alignment charts for calculations relating to moist air.

He shows that all principal equations relating to moist air are of the general form:

$$f_3(c) = f_1(a) \,/\, f_2(b)$$

and that such equations can be represented by alignment charts consisting of two parallel straight lines intersected by a third straight line.

The author explains in detail how alignment charts can be constructed which show: (1) The relation of total heat to moisture content and temperature. (2) The relation of relative humidity to moisture content, temperature, and pressure. (3) The relation of specific volume and specific weight to pressure, temperature, and total heat. He includes seven large charts for use in such calculations.

The subject has been treated in a very thorough and complete manner. The charts are new and are based on the latest available physical constants. Their use is explained by means of 18 numerical examples.

Arc Welding Design and Practice

["Procedure Handbook of Arc Welding Design and Practice," 5th ed. 1938. 1012 + xii pp., 5¾ x 9 in., semi-flexible binding. Published by the Lincoln Electric Co., 12818 Coit Rd., Cleveland, Ohio. Price, $1.50 in U. S. A., $2.00 elsewhere.]

Re-issued each year to include new data essential for efficient use of arc welding and its varied applications, the new edition of this well known handbook is encyclopedic in scope, concisely written and profusely illustrated, with over 1200 photographs and drawings. It was prepared especially for the use of designers, engineers, architects, production managers, welding supervisors and operators, and has much of interest for draftsmen, steel fabricators and erectors, maintenance managers and piping and pipe line contractors.

The eight sections of the book cover welding methods and equipment; technique of welding; procedures, speeds and costs for welding mild steel; structure and properties of weld metal; weldability of metals; designing for arc welded steel construction of machinery; designing for arc welded structures; and typical applications of arc welding in manufacturing, construction and maintenance.

In the section on typical applications, over 40 pages are devoted specifically to welding piping, including high temperature piping.

Furnaces and Unit Heaters

["Air Conditioning—Furnaces and Unit Heaters," by J. Ralph Dalzell, head, architectural engineering department, American School. 1st ed. 1938. 430 + xiv pp., 5½ x 8¼ in., clothbound. Published by American Technical Society, Drexel Ave. at 58th St., Chicago, Ill. Price, $3.00.]

This new book is intended as a text and reference on the application of gravity and mechanical furnaces, and unit heaters for commercial and domestic heating.

Chapters are included on principles of ventilation, transmission coefficients and tables, heating and cooling loads, insulation, air conditioning principles, gravity furnaces, mechanical warm air furnaces, air conditioning furnaces, registers and grilles, electric heating, humidification, unit heaters and automatic controls.

A. S. T. M. Publishes Proceedings

The 1937 "Proceedings" of the American Society for Testing Materials are published in two parts, Part I comprising committee reports (with appended papers) and new and revised A. S. T. M. tentative standards. Part II gives all technical papers, including the Marburg lecture on plastics.

In Part I are 45 committee reports outlining important standardization and research work. This part also includes some 118 specifications and tests prepared for the first time in 1937 or revised. Nine reports involve ferrous metals, including steel, yield point of structural steel (final report), wrought iron, cast iron, corrosion, malleable iron castings, iron-chromium-nickel and related alloys, fatigue of metals and effect of temperature on properties of metals. The latter has appended items covering discrepancies in load carrying abilities of carbon steels at 850 F, and long time creep tests on certain steels. The presidential address by Dr. A. C. Fieldner on "Fuels of Today and Tomorrow" is also included.

Forty-seven technical papers are included in Part II. Six papers comprise the symposium on significance of tests of coal. Several relate to metallic materials, including weld metal, several on cast iron, analysis of the Brinell hardness test, and fatigue properties of non-ferrous sheet metals. Three papers pertain to the use of metals at elevated temperatures. There are also papers on water, particularly boiler feedwaters.

Parts I and II can be purchased at $5.50 each in stiff paper cover; $6, cloth; and $7, half-leather—from A. S. T. M. Headquarters, 260 S. Broad St., Philadelphia, Pa.

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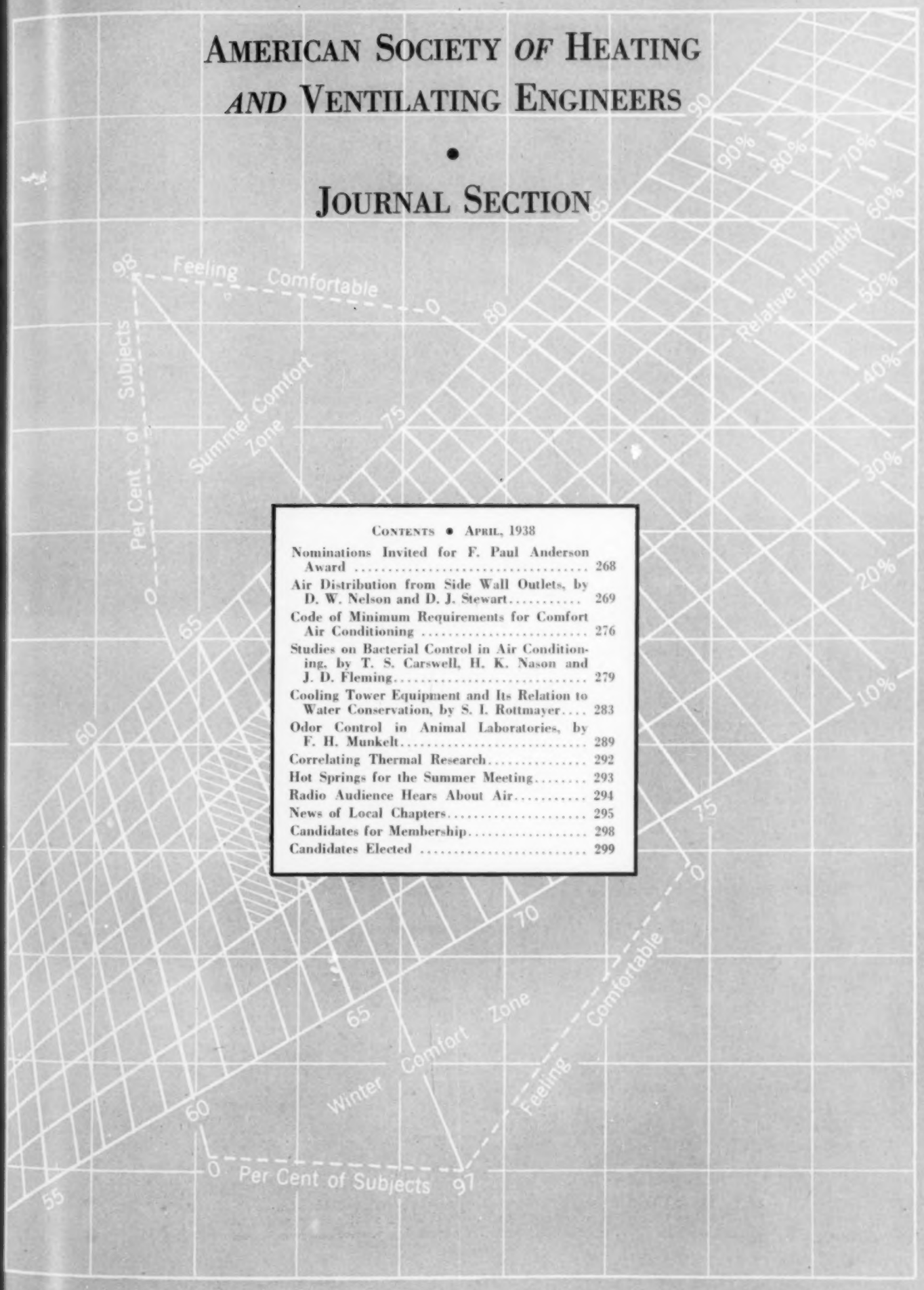

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IOWA - NEBRASKA: *Organized,* 1937. *Headquarters,* Omaha, Neb. *President,* A. L. WALTERS, 900 29th St., Des Moines, Ia. *Secretary,* W. R. WHITE, 4339 Larimore Ave., Omaha, Neb.

KANSAS CITY: *Organized,* 1917. *Headquarters,* Kansas City, Mo. *Meets,* Second Monday in Month. *President,* A. H. SLUSS, 827 Mississippi Ave., Lawrence, Kan. *Secretary,* GUSTAV NOTTBERG, 914 Campbell St.

MANITOBA: *Organized,* 1935. *Headquarters,* Winnipeg. Man. *Meets,* Fourth Thursday. *President,* D. F. MICHIE, 492 Wardlaw Ave. *Secretary,* E. J. ARGUE, Ste. 23, Estelle Apts.

MASSACHUSETTS: *Organized,* 1912. *Headquarters,* Boston, Mass. *Meets,* Third Tuesday in Month. *President,* JAMES HOLT, Massachusetts Institute of Technology, Cambridge, Mass. *Secretary,* H. C. MOORE, 69 Massachusetts Ave., Cambridge, Mass.

MICHIGAN: *Organized,* 1916. *Headquarters,* Detroit, Mich. *Meets,* First Monday after the 10th of the Month. *President,* F. J. FEELY, 950 Trombley Rd., Grosse Pointe Pk. *Secretary,* G. H. TUTTLE, 2000 Second Ave.

WESTERN MICHIGAN: *Organized,* 1931. *Headquarters,* Grand Rapids, Mich. *Meets,* Second Monday in Month. *President,* W. W. BRADFIELD, 341 Michigan Trust Bldg. *Secretary,* S. W. TODD, JR., 309 Paris, S. E.

MINNESOTA: *Organized,* 1918. *Headquarters,* Minneapolis, Minn. *Meets,* Second Monday in Month. *President,* R. E. BACKSTROM, Room 1981, First Natl. Bank Bldg., St. Paul, Minn. *Secretary,* F. C. WINTERER, 836 Juno St., St. Paul, Minn.

MONTREAL: *Organized,* 1936. *Headquarters,* Montreal, Que. *Meets,* Third Monday. *President,* G. L. WIGGS, University Tower. *Secretary,* C. W. JOHNSON, 630 Dorchester St., W.

NEW YORK: *Organized,* 1911. *Headquarters,* New York, N. Y. *Meets,* Third Monday in Month. *President,* W. E. HEIBEL, 11 West 42nd St. *Secretary,* T. W. REYNOLDS, 100 Pinecrest Dr., Hastings-on-Hudson, N. Y.

WESTERN NEW YORK: *Organized,* 1919. *Headquarters,* Buffalo, N. Y. *Meets,* Second Monday in Month. *President,* B. C. CANDEE, 19 Tremont Ave., Kenmore, N. Y. *Secretary,* W. R. HEATH, 119 Wingate Ave.

NORTHERN OHIO: *Organized,* 1916. *Headquarters,* Cleveland, O. *Meets,* Second Thursday in Month. *President,* PHILIP COHEN, 401 East Ohio Gas Bldg. *Secretary,* C. A. MCKEEMAN, Case School of Applied Science.

OKLAHOMA: *Organized,* 1935. *Headquarters,* Oklahoma City, Okla. *Meets,* Second Monday. *President,* E. F. DAWSON, University of Oklahoma, Norman, Okla. *Secretary,* E. W. GRAY, Box 1498, Oklahoma City, Okla.

ONTARIO: *Organized,* 1922. *Headquarters,* Toronto, Ont. *Meets,* First Monday in Month. *President,* G. A. PLAYFAIR, 113 Simcoe St. *Secretary,* H. R. ROTH, 57 Bloor St. W.

PACIFIC NORTHWEST: *Organized,* 1928. *Headquarters,* Seattle, Wash. *Meets,* Second Tuesday in Month. *President,* W. W. COX, 326 Columbia St. *Secretary,* M. N. MUSGRAVE, 314-9th Ave., N.

PHILADELPHIA: *Organized,* 1916. *Headquarters,* Philadelphia, Pa. *Meets,* Second Thursday in Month. *President,* L. P. HYNES, 240 Cherry St. *Secretary,* C. B. EASTMAN, 530 Brookview Lane, Brookline, Upper Darby, Pa.

PITTSBURGH: *Organized,* 1919. *Headquarters,* Pittsburgh, Pa. *Meets,* Second Monday in Month. *President,* J. F. COLLINS, JR., 435 Sixth Ave. *Secretary,* T. F. ROCKWELL, Carnegie Inst. Tech.

ST. LOUIS: *Organized,* 1918. *Headquarters,* St. Louis, Mo. *Meets,* First Tuesday in Month. *President,* G. W. F. MYERS, 3947 W. Pine Blvd. *Secretary,* D. J. FAGIN, 1344 Woodruff Ave.

SOUTHERN CALIFORNIA: *Organized,* 1930. *Headquarters,* Los Angeles, Calif. *Meets,* Second Tuesday in Month. *President,* E. H. KENDALL, 1978 S. Los Angeles St. *Secretary,* J. F. PARK, 1234 South Grand.

TEXAS: *Organized,* 1936. *Headquarters,* College Station, Texas. *President,* R. F. TAYLOR, 909 Banker's Mortgage Bldg., Houston, Tex. *Secretary,* W. H. BADGETT, Texas Engrg. Experiment Station, College Station, Tex.

WASHINGTON, D. C.: *Organized,* 1935. *Headquarters,* Washington, D. C. *Meets,* Second Wednesday in Month. *President,* L. OURUSOFF, 411 Tenth St., N. W. *Secretary,* L. F. NORDINE, Room 203, 734 Jackson Pl., N. W.

WISCONSIN: *Organized,* 1922. *Headquarters,* Milwaukee, Wis. *Meets,* Third Monday in Month. *President,* J. H. VOLK, 1906 W. St. Paul Ave. *Secretary,* H. C. FRENTZEL, 3000 W. Montana St.

Nominations Invited for F. Paul Anderson Award

The Committee on the F. Paul Anderson Medal Award invites all members of the Society to submit the names of candidates for the third award of the F. Paul Anderson Medal.

Chairman J. F. McIntire has outlined briefly the conditions under which the award is to be made as follows:

(1) Any Member of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in good standing during 1938 is eligible for nomination.

(2) Each nomination sent to the Committee on Award must be accompanied by a statement of the candidate's accomplishments, specifically mentioning the work done or services performed and indicating in what way such work or services have been outstanding in the field of heating, ventilating or air conditioning, during any period prior to 1938.

(3) All nominations must be in the hands of the Committee on July 1, 1938, and are to be addressed to the Committee on F. Paul Anderson Medal Award, c/o Secretary, AMERICAN SOCIETY OF HEATING & VENTILATING ENGINEERS, 51 Madison Ave., New York, N. Y.

At the 44th Annual Meeting in New York Pres. E. Holt Gurney announced the personnel of the Committee as follows: J. F. McIntire, *Chairman,* Prof. E. O. Eastwood, W. T. Jones, Prof. A. P. Kratz, and Prof. G. L. Larson.

Under the regulations governing the F. Paul Anderson Award, the Committee will report to the Council at its last meeting of the calendar year and the presentation of the medal will be scheduled for the Annual Meeting.

The F. Paul Anderson Award was created and announced at the 36th Annual Meeting by Pres. Thornton Lewis and the first presentation of the medal was made to W. H. Carrier at the 38th Annual Meeting in Cleveland, January 1932. The second presentation was made to Dr. A. C. Willard at the 42nd Annual Meeting in Chicago, January 1936.

The medal is named for the late F. Paul Anderson, Past President of the Society, a former Director of the Research Laboratory, and for over 40 years Dean of the College of Engineering at the University of Kentucky. The F. Paul Anderson Medal was designed by John Swanson and carries a likeness of Dean Anderson on one side and on the reverse side figures typifying heating, ventilating and air conditioning.

The Committee on Award will welcome the cooperation of members of the Society and suggests that nominations be sent promptly.

Air Distribution from Side Wall Outlets

By D. W. Nelson* (*MEMBER*), Madison, Wis., and D. J. Stewart** (*MEMBER*), Rockford, Ill.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with the University of Wisconsin, conducted under the supervision of the Research Technical Advisory Committee on Air Distribution.

THE selection of outlets, which will deliver sufficient air for effectively cooling a space and accomplish this within the limits of air motion and temperature required for comfort, often presents a difficult problem to the engineer. Although there are other satisfactory methods of supplying air to a room, this paper will discuss only the problems in connection with the introduction of air through side-wall openings.

The Practical Problem

The engineer who selects the air distribution outlets usually has the following information to guide him: (1) The room dimensions are known; (2) A temperature differential between the entering air and the control temperature of the room has been selected; (3) The air volume required for supplying the necessary cooling has been computed; (4) The maximum velocity permissible from the standpoint of noise has been calculated; (5) The desired grille mounting height is known, either exactly or within rather close limits; (6) The desired minimum room temperature is also known.

The problem is to find the smallest outlet, or outlets (bearing in mind the noise limitation on velocity) which will give satisfactory air distribution. The range of selection is often limited and the entire problem complicated by additional considerations such as beamed ceilings, large posts or other obstructions, non-rectangular rooms, and the location of available space for the outlet itself or the ducts leading to it.

It is often possible to find two or more arrangements of size and location of outlets which will be satisfactory. Generally, the one involving the smallest total outlet area is the most satisfactory.

The Criteria for Comfort

The necessity for some standard of satisfactory air distribution is apparent. The literature[1] on air distribution is meager and, as none of the papers deal primarily with this subject, it was necessary to make some study from which suitable criteria could be established. That work will not be discussed here except to say that the observations were limited and the subjects few and not sufficiently diversified as to types of people. Nevertheless, these observations, together with the data in the literature referred to, supplemented by considerable field experience, were used to establish a standard test procedure to determine whether or not the distribution in a given room under a given set of conditions was or was not satisfactory.

The accompanying chart of Fig. 1 is a family of curves, each curve representing the maximum permissible difference in room temperature for various air velocities. All combinations below the line are considered to be unsatisfactory, and those above the line satisfactory. It will be noted that, with higher room temperatures, a higher velocity is permitted with the same air temperature difference, or a greater temperature differential is permitted with the same velocity. The velocities referred to are average velocities as measured with a Katathermometer. The temperature difference referred to is in every case the difference between the temperature of the air at the 5-ft level directly below the inlet and the temperature at any location where the air velocity is measured. All temperatures are measured with exposed thermocouples. It will be observed that this chart is in substantial agreement with what data there are on this subject in the literature referred to.

Theoretically, the establishment of the fact that any room meets the requirements as shown by this chart involves the necessity of measuring the air velocity and temperature difference at every point in the occupied zone. However, with experience one can locate the worst places by inspection, combined with approximate readings of temperature and velocity, so that it is rarely necessary to make more than a few accurate measurements.

Since the requirement stated only sets a high limit, it does not take care of the possibility of improperly cooling certain sections of the room. One solution is to measure temperature rise, but, except near places where there is heat gain, this is not satisfactory, for the

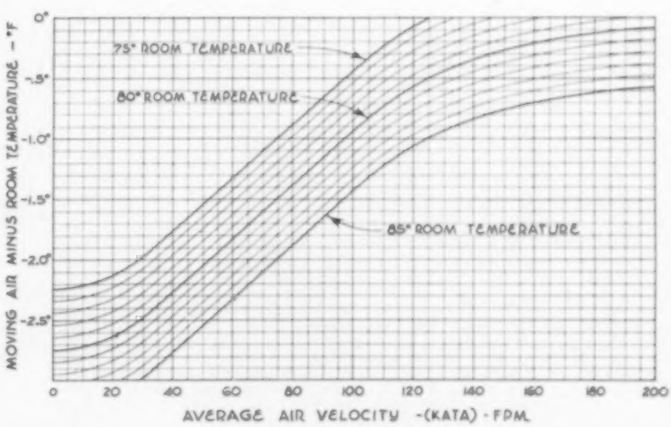

Fig. 1—Relation between maximum air velocity and moving air temperature for various room temperatures

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[1] Cooling Requirements for Summer Comfort Air Conditioning, by F. C. Houghten, F. E. Giesecke, C. Tasker, and C. Gutberlet (A. S. H. V. E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, Dec. 1936, pp. 681-692).

A. S. H. V. E. GUIDE, 1937, Chapter 8, p. 76.

Classroom Drafts in Relation to Entering Air Stream Temperature, by F. C. Houghten, H. H. Trimble, C. Gutberlet, and M. F. Lichtenfels (A. S. H. V. E. TRANSACTIONS, 1935, Vol. 41, pp. 268-278).

Presented at the 44th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, at the Joint Session with the *N. W. A. H. & A. C. A.*, New York, N. Y., January, 1938.

Fig. 2—The test room looking toward the air discharge outlet

temperature rise will be negligibly small, yet when the space is in use the occupants may increase the temperature and relative humidity to an extent which may feel uncomfortable. A chart similar to that shown but specifying *minimum* velocities and air temperature differences is desired, but again one must measure the air velocity and temperature at all points in the room and must make allowance for the heat gain in that region. In an odd-shaped room, it is often difficult to determine by inspection where the worst places will occur. However, in a square or rectangular room with outlets and return grilles on the same wall and without greatly localized heat sources other than occupants, experience shows that it is usually only necessary to know that sufficient cooling air will reach the far end of the room. The air discharged into the room, in finding its way back to the return grille, necessarily passes below the entering air. The large cross-section available for the return air insures that the velocities are low, and the symmetry of the rectangular room insures that the cooling air will be well distributed.

Even for this simple room there is no single value for the velocity at the rear wall which will satisfy all cases. For example, if the rear wall is a large source of heat gain, the velocity down it should be considerably greater than if it is thoroughly insulated and therefore a source of little or no heat gain. Furthermore, the proper velocity depends to some extent upon the type of occupancy. However, experience indicates that an average velocity of 30 fpm is about the minimum that can be permitted down the back wall. A slightly lower value will be satisfactory if there will be no heat gain from that wall.

The maximum permissible velocity down the back wall is controlled by considerations of the discomfort produced upon occupants of the room seated near that wall. The proper method of determining this maximum velocity would be in accordance with the chart, Fig. 1. The temperature difference of the air falling down the back wall will, in the case of most outlets which are otherwise satisfactory, rarely exceed 1 F. Accordingly, the maximum permissible velocity, if the room temperature were 80, would be 100 fpm; and if the room temperature were 75, would be 75 fpm. It is well to keep this particular velocity considerably on the low side, since air which comes down the back wall is deflected at the floor and returns along the floor where the occupant is particularly sensitive to a feeling of discomfort from a low temperature. Accordingly, an arbitrary requirement was made that the average velocity down the back wall should not exceed 75 fpm.

There is one other uncomfortable condition not mentioned thus far and not covered by Fig. 1. Regardless of the air temperature, there is some air velocity, even if momentary, which is uncomfortable if only because it disturbs papers and other light articles or the occupant's hair. This maximum acceptable velocity is believed to be in the neighborhood of 200 fpm, and accordingly one requirement for a satisfactory outlet is that the maximum air velocity within the zone of occupancy shall not be more than 200 fpm as measured with an instrument which indicates approximately the momentary velocity.

The Test Room and Instruments

The room in which the data for this article were taken was constructed particularly for air distribution tests. Fig. 2 is a photograph taken within this room, and Fig. 3 is a diagrammatic plan. It has been partially described in a paper on grille noise,[2] but the description is repeated here to obviate the necessity of referring to the other paper. The building itself is of poured concrete, pillar and slab construction, with a flat concrete ceiling covered with built-up composition roofing. The test room is on the sixth (top) floor of this building, and is 17 ft wide and 9 ft 7 in. high. The inner side walls are of sheet metal, outside of which are narrow spaces, heated to maintain room temperature during tests with cool air. The ceiling is ¼ in. rigid insulation on joists about a foot below the concrete slab of the roof. The floor is concrete. An exhaust grille, extra large to reduce velocity to a negligible value, is located in the center of one wall at the floor. This wall is also of ¼ in. rigid insulation and forms the front of a pressure box 5 ft deep and 8 ft wide extending from the top of the exhaust grille to the ceiling. Air is introduced into this box near the bottom and at velocities greatly reduced by a diverging nozzle with splitters. The front of the pressure box is in sections so arranged that a grille of any size and shape can be installed at any height desired. Behind the test grille is mounted a 3-ft section

[2]The Noise Characteristics of Air Supply Outlets, by D. J. Stewart and G. F. Drake (A. S. H. V. E. Journal Section, *Heating, Piping and Air Conditioning*, Jan. 1937, pp. 65-70).

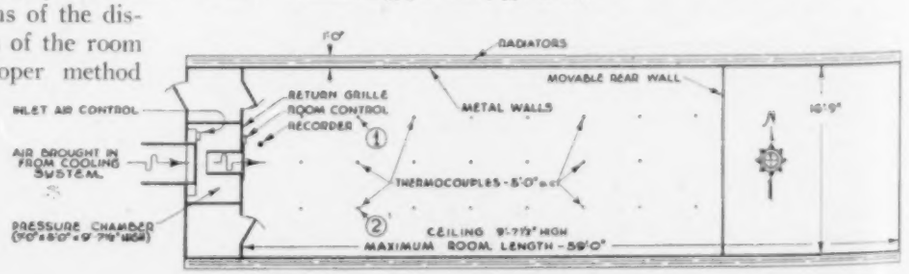

Fig. 3—Diagrammatic plan of test room and pressure box

of duct having the nominal dimension of the grille to simulate an actual installation. Careful tests have indicated that the flow of air from the pressure box through this duct is satisfactorily uniform over its entire cross-section.

The room is heated by a panel system and the room temperature is maintained constant by a control system governing the temperature of the air behind the panel, supplemented with a pilot thermostat in the room. Since very small temperature differences are important in determining whether or not the particular grille meets the velocity and temperature requirements, it is necessary that the room temperature be held within very close limits during the period of one test. A room temperature change of as much as 0.5 F during one set of readings necessitates repetition of the test.

The temperature of the entering air does not need to be so accurately controlled. A temperature variation of as much as 1 F during a single test will not appreciably affect the readings. A standard proportioning control is used in the blast system supplying the air. Frequent thermocouple temperature readings are taken of return and entering air temperatures.

Thermocouples are installed about the room on 5 ft centers horizontally and 2½ ft centers vertically, as indicated in Fig. 3. A number of these are also shown in the photograph of Fig. 2.

The temperature differences were measured directly by the expedient of locating the return junction of the thermocouple galvanometer system in the aspirated air. The return junction thermocouple location actually used for all the readings shown was at the 5 ft level about 1 ft from the wall and directly beneath the air outlet.

The measuring circuit is arranged so that a sensitive galvanometer determines the balancing potential necessary for zero current in the thermocouple circuit, and a potential multiplying system enables the net voltage of the two junctions to be read directly. This voltage is a measure of the difference in temperature at the two points and the voltmeter can be calibrated directly in degrees Fahrenheit. By this means temperature differences were measured with an accuracy of approximately ±0.1 F between readings. The absolute value of the differences is not as accurately known as this, but the error, if any, is also in the zero temperature line of Fig. 1.

A recording thermometer of a type having a current of air over the thermostatic element was used to record the room temperature at a point adjacent to the return junction of the thermocouple. Records of room temperature refer to the readings of this instrument. Measurements of the inlet and exhaust air temperatures were made with thermocouples and a commercial potentiometer having its return junction compensated for the ambient temperature. This instrument, as well as the fan recorder, has an accuracy of approximately ±0.5 F.

The volume of air discharged was measured with a Wahlen gage and a carefully calibrated Venturi meter located in the duct supplying the pressure box. The leakage from the pressure box at various differences in pressure between it and the room was obtained by measurement of the flow through the Venturi meter with the grille opening closed. A slope gage indicates the static pressure in the box and the leakage calculated from this pressure was deducted from all Venturi meter readings in determining the total volume discharged into the room.

The outlet face velocity was obtained by dividing the volume as determined by the Venturi meter by the *core area* of the grille or opening. By *core area* is meant the actual opening inside of the grille frame in cases where a grille was used, and the area of the opening where no grille was used. As a rough check against the possibility of a gross error in calculation, the face velocity was in each instance measured with a deflecting vane type indicating instrument.

Room velocities were measured either with the deflecting vane type indicating instrument or with a Kata-thermometer. High temperature Kata-thermometers were used, two or more consecutive readings being averaged for each record. The air temperature was in every case measured by a thermocouple at a point adjacent to the Kata-thermometer and at the time the Kata reading was taken. The velocities shown on the charts with this article were taken with a deflecting vane instrument and represent mental averages in one direction (what might be called the vector average)—not the non-directional (or scalar) average as measured by the Kata-thermometer.

In order to provide for rooms of various lengths, the rear wall of the room is movable, as shown in Fig. 3. The thermocouple drops are inserted into plugs in the ceiling so that they can be removed when the length of the room is to be altered.

The Test Data

The results of one series of tests made in this room are shown in Figs. 4 to 8, inclusive. In this series used for illustrative purposes, five open end ducts and five grilles to fit them were used. The aspect ratio of the grilles varied from 1.00 to 12.2, with the areas maintained fairly constant. All grilles were of one type, with thin horizontal closely spaced guide vanes giving a slight upward deflection to the air stream. In the particular series shown, an entrance velocity of 1000 fpm and a 20 F temperature difference were used throughout; the length of room was fixed at 40 ft, and the grille mounting height was 7 ft 6 in. Each of the figures has two parts, one marked *A* for the test with the open end duct and the other *B* for the test with the grille fitted in place.

Temperature differences were measured with the thermocouples at 120 positions at 5 ft spacings and at five elevations. The 40 readings taken along the vertical center plane of the room in line with the grille are shown on Figs. 4 to 8 as differences between the observed temperature and the room temperature. The edges of the primary stream were first located and marked at each 5 ft location by the use of cigarette smoke, and checked by observing the path of a large quantity of smoke introduced with the air supply. This air stream path is shown on the figures as a broken line. In the air stream the average velocity was determined at elevation intervals of a few inches at each of the 5 ft locations. At each one of 35 thermocouple locations, the average and maximum velocities were taken. In all, about 125 such velocity readings were made in each test. For this purpose a deflecting vane type meter was used and the determinations of the average velocities were used to plot lines of equal velocities on the figures. In addition, Kata-thermometer readings were made at certain points in the room. Some of the pertinent details of these tests are summarized in Tables 1 and 2.

Table 1—Performance Characteristics for Air Discharged from an Open Hole
(Room Length, 40 ft; Face Velocity, 1000 fpm)

1		Nominal Size	11x11	16x8	19x7	28x5	36x4
2		Aspect Ratio	1.00	2.03	2.78	5.84	9.54
3		Frame Area (sq ft)	0.803	0.847	0.879	0.915	0.920
4		Frame Area/Duct Area	0.956	0.954	0.950	0.940	0.920
5		Supply Volume (cfm)	803	847	879	915	920
6		Room Temperature, deg F	80	80	79	81	82
7		Supply Temp. Diff., deg F	—20	—20	—19	—20	—20
8	Data at 5 ft level	Max. Peak Velocity[a], fpm	300	250	250	300	350
9		and Location, feet	20	15	20	20	15
10		Av. Vel. Same Loc.[a], fpm	225	200	200	200	300
11		Max. Vel. on CL (Kata), fpm	130	170	142	245	208
12		Location, feet	25	20	20	19	20
13		Temp. Diff., deg F	—2.5	—3.3	—3.1	—4.0	—3.4
14		Velocity at (1)[b] (Kata), fpm	40	40	20	25	45
15		Temp. Diff., deg F	—0.5	+1.0	+1.5	0	—0.5
16		Velocity at (2)[b] (Kata), fpm	25	20	38	20	30
17		Temp. Diff., deg F	—0.5	+1.0	+1.5	0	0
18		Vel. at Rear Wall (Kata), fpm	100 up	60 up	75 up	90 up	88 up
19		Temp. Diff., deg F	—1.7	—1.0	—0.8	—2.5	—0.9
20		Distance from Opening, feet Stream Hits Floor	28	25	27	23	28
21		Distance from Opening, feet Primary Jet Drops to 150 fpm	25	26	26	27	23

[a]Data with deflecting vane type velocity meter.
[b]See Fig. 3 for location.

Lines of equal velocity were plotted on Figs. 4 to 8, using the large number of readings obtained by the deflecting vane instrument, which made it possible to trace out the major air movements in the room. Naturally, there were many miscellaneous movements or turbulences that can hardly be recorded or plotted. The averages were obtained by observing the swing of the indicating hand of the instrument. Since this type of instrument is directional, it is necessary to hold it into the direction from which the air is coming. This is difficult to do under turbulent conditions and the visual averaging of the swing of the hand also adds to the difficulty of obtaining much accuracy in these determinations. Four different instruments of this type were available and were found to check very closely among themselves, and are considered to be satisfactory for tracing the air currents as plotted on these charts. It should be borne in mind that the values shown in Figs. 4 to 8 are directional and should not be used in interpreting comfort conditions in connection with Fig. 1. The velocities shown on Fig. 1 are non-directional as determined by a Kata-thermometer. Velocity readings were taken at various points in the room with Kata-thermometers as recorded in Tables 1 and 2. The four Kata instruments available gave consistent results.

The temperature differences obtained in the center plane of the room are shown on the charts on the 5 ft vertical lines in the locations in which they were observed. For clarity, the velocity determinations are placed on the intermediate vertical lines. The method used in taking and in plotting velocities and temperature differences is considered to result in a good picture of the flow pattern of the various openings and the related temperature differences.

The items in Tables 1 and 2 are perhaps self-explanatory, although attention might be called to a few of them. For the open hole tests the maximum peak velocities were found to be from 250 to 350 fpm at a distance 15 to 20 ft from the supply outlet. The average velocity at the same locations was found to be 200 to 300 fpm. In the case of the grille tests, the maximum peak velocities varied from 180 to 230 fpm at a distance of 25 to 30 ft out, and the average velocities were 75 to 140 fpm at the same location. These figures are given in items 8, 9 and 10 of Tables 1 and 2. These particular figures do not clearly indicate any effect due to aspect ratio. As pointed out elsewhere, the velocities measured with an indicating vane meter are useful for the purpose of obtaining a general picture of the flow pattern, but are not accurate when the stream is very turbulent.

Items 11 to 19 of these tables concern comfort conditions and may be used in connection with standards such as shown in Fig. 1 to indicate satisfactory or unsatisfactory performance. The highest velocities, as determined by the Kata-thermometer, and temperature differences found at the 5 ft level indicate that unsatisfactory combinations existed in all of the open hole tests. These velocities range from 130 to 250 fpm, and the temperature differences from —2.5 F to —4 F. In the grille tests the three with low aspect ratios showed satisfactory performance at all distances at the 5 ft level, as judged by standards of Fig. 1. The two with high aspect ratios showed higher velocities, with greater temperature differences at locations nearer the supply outlet than did the other three grilles. Reference to Fig. 1 indicates unsatisfactory performance for these two. An increase in outlet velocity might remedy this unsatisfactory performance, but without an increased length of room might result in an excessive velocity down the rear wall. The maximum Kata-thermometer velocities for all grilles range from 55 to 150 fpm, and the temperature differences from —0.7 F to —3 F.

The air movement in the return air stream flowing towards the return grille is indicated in items 14 to 17. For the open hole tests, the average velocity at points 1 and 2 on Fig. 3 was 30 fpm, and the average temperature difference was between +0.3 and +0.4 F. For the grille test, the corresponding average velocity was 23 fpm at an average temperature difference near

Table 2—Performance Characteristics for Air Discharged from a Grille
(Room Length, 40 ft; Face Velocity, 1000 fpm)

1		Nominal Size	11x11	16x8	19x7	28x5	36x4
2		Aspect Ratio	1.00	2.17	3.04	6.94	12.2
3		Core Area (sq ft)	0.677	0.711	0.729	0.723	0.696
4		Core Area/Duct Area	0.806	0.800	0.790	0.744	0.696
5		Supply Volume (cfm)	677	711	729	723	696
6		Room Temperature, deg F	81	80	81	80.5	81.5
7		Supply Temp. Diff., deg F	—20.5	—20	—21	—21	—20
8	Data at 5 ft level	Max. Peak Velocity[a], fpm	185	180	180	225	230
9		and Location, feet	25	30	30	25	25
10		Av. Vel. Same Loc.[a], fpm	125	75	100	140	100
11		Max. Vel. on CL (Kata), fpm	70	70	55	150	120
12		Location, feet	28	30	30	23	21
13		Temp. Diff., deg F	—0.7	—1.5	—1.3	—3.0	—1.3
14		Velocity at (1)[b] (Kata), fpm	28	24	20	34	15
15		Temp. Diff., deg F	0	+1.0	—0.5	+0.5	—0.5
16		Velocity at (2)[b] (Kata), fpm	20	20	10	36	25
17		Temp. Diff., deg F	0	+1.0	—0.5	+0.5	—0.7
18		Vel. at Rear Wall (Kata), fpm	120 dn.	120 dn.	55 dn.	50 tur.	55 tur.
19		Temp. Diff., deg F	—2.1	—1.6	—1.2	—1.8	—1.0
20		Distance from Grille, feet Stream Hits Floor	40+	40+	37[c]	40[c]	40[c]
21		Distance from Grille, feet Primary Jet Drops to 150 fpm	40+	40+	35	34	28

[a]Data with deflecting vane type velocity meter.
[b]See Fig. 3 for location.
[c]Actual stream diffuses before reaching floor. Distance given is where lower stream edge extended would strike.

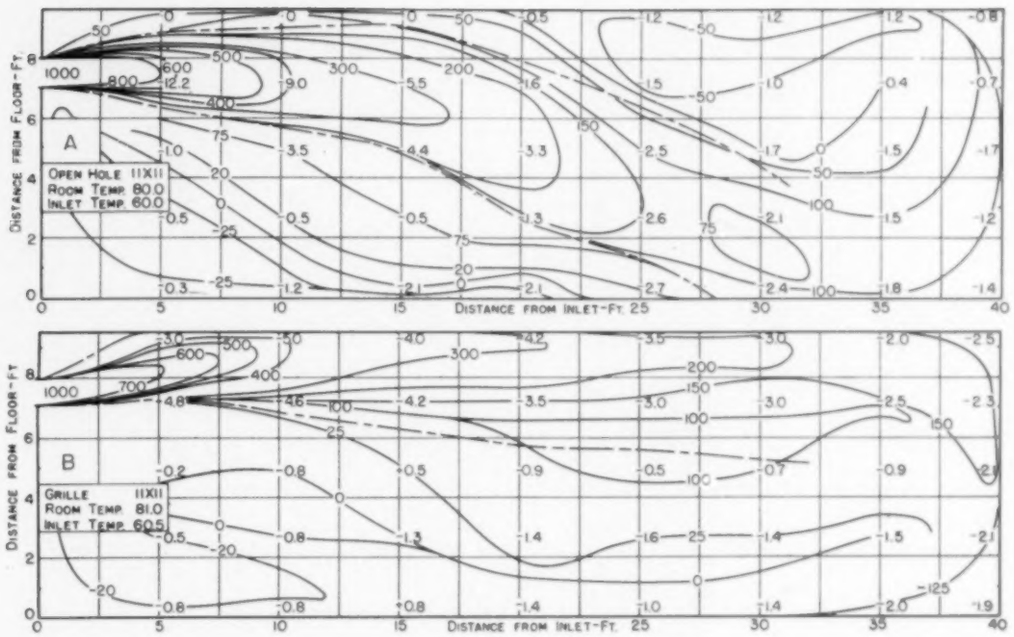

Fig. 4—Air distribution along vertical section through center of room: *A* (*above*) for 11 x 11 open hole; and *B* (*below*) for 11 x 11 grille

+0.1 F. The difference in velocities is about in the same proportion as the ratio of average volumes in item 5 of the two tables. The slight difference in temperature differences is probably of no significance.

In each of the open hole tests, the velocity at the back wall is upward and, as far as the standards of Fig. 1 are concerned, is on the border line of acceptability, but actually is not to be tolerated, as the upward flow indicates falling short. For the first two grilles tested, the flow is definitely down and the velocity somewhat beyond the acceptable limit as indicated in Fig. 1. The air stream is therefore reaching the rear wall at too high a velocity. The performance probably would be satisfactory in a slightly longer room. For the 40 ft room a slightly lower velocity probably would result in satisfactory performance. The third grille, 19 x 7, had an acceptable velocity down the rear wall and, in fact, is acceptable in every respect in accordance with the requirements set forth in this paper. The two remaining grilles with high aspect ratios had low velocities at the rear wall. These velocities were rather mixed, although apparently more upward than downward. These two grilles, however, as shown in item 11, have excessive velocities at the 5 ft level.

Items 20 and 21 showing velocities taken with the deflecting vane type meter indicate the length of throw of the various air streams. With the open hole, little or no effect is shown on the throw with change in aspect ratio. With the grilles, some tendency to a shortened throw with high aspect ratio is indicated, although this is somewhat obscured by the influence of diffusion. In the open hole test, the stream hit the floor at

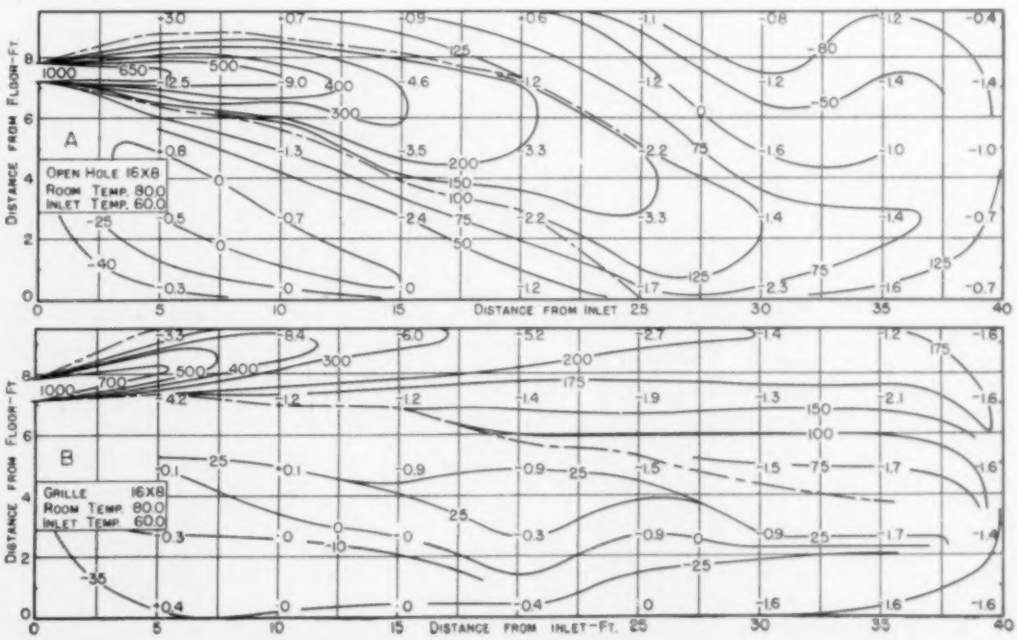

Fig. 5—Air distribution along vertical section through center of room: *A* (*above*) for 16 x 8 open hole; and *B* (*below*) for 16 x 8 grille

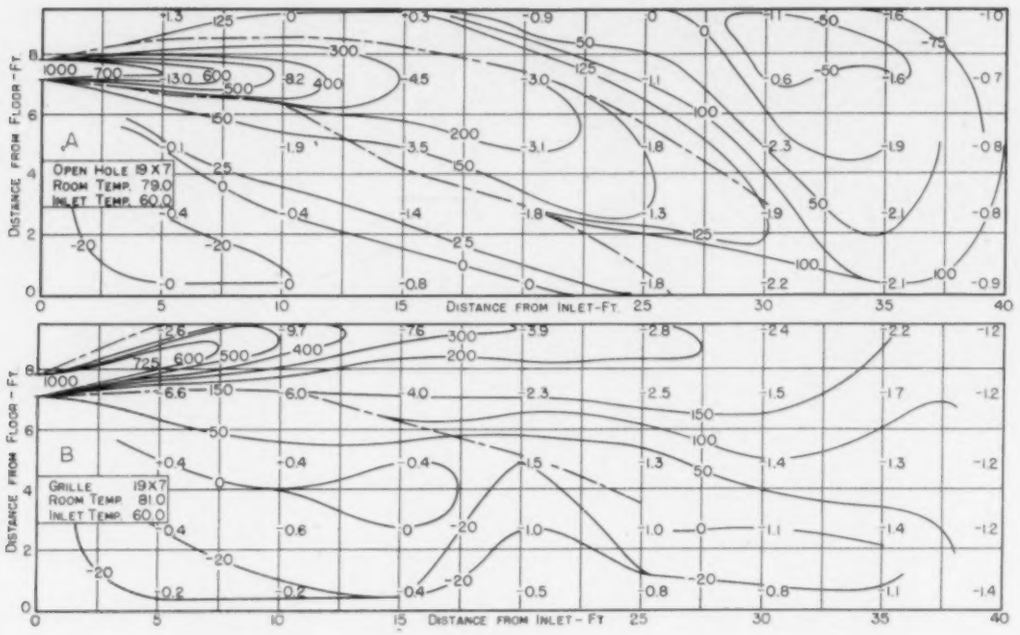

Fig. 6—Air distribution along vertical section through center of room: *A* (*above*) for 19 x 7 open hole; and *B* (*below*) for 19 x 7 grille

distances from 23 to 28 ft from the supply outlet. When the grilles were in place, because of diffusion, the stream could hardly be said to hit the floor at all, but a continuation of the lower edge of it for the three higher aspect ratios would strike the floor between 37 ft and the end of the room.

The location where the air stream is reduced to an average velocity of 150 fpm, as indicated by a deflecting vane type meter, is shown in item 21 of the tables. For the open hole tests, no influence is indicated of the effect of the aspect ratio. The air stream reaches the rear wall with this velocity in the tests of the low aspect ratio grilles. In the grille with the highest aspect ratio this velocity was reached at 28 ft out. The average distance measured horizontally for the reduction of the air stream velocity to 150 fpm was 25 ft for the open hole tests and 36 ft for the grille tests.

Factors Affecting the Distribution of Air

A quantitative discussion of the proper outlet to use for even a limited number of representative rooms is much beyond the scope of this paper, but it is possible to point out in a qualitative manner the effects of some of the significant factors involved. The following statements are made as a result of a large number of experiments of the sort described in this paper, supplemented by experience with actual installations. A list of some of the principal variables other than throw (usually the length of the room) follows:

(1) Height of room; (2) Mounting height of grille; (3) Volume of air; (4) Temperature of air; (5) Velocity of air; (6) Aspect ratio; (7) Side walls; (8) Location of outlet; (9) Beams.

The first two of these are interrelated to a consider-

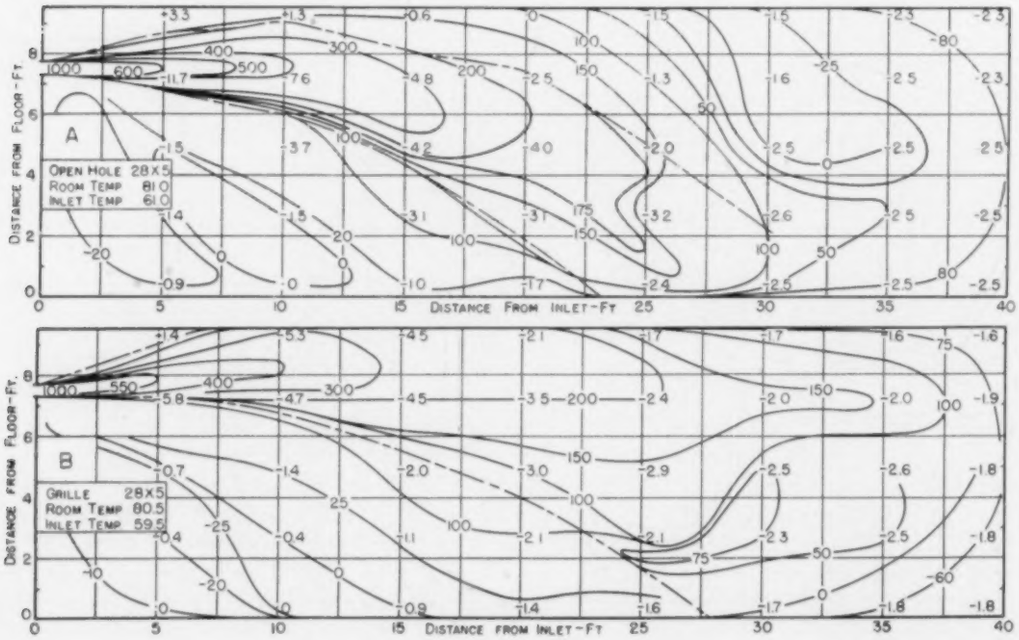

Fig. 7—Air distribution along vertical section through center of room: *A* (*above*) for 28 x 5 open hole; and *B* (*below*) for 28 x 5 grille

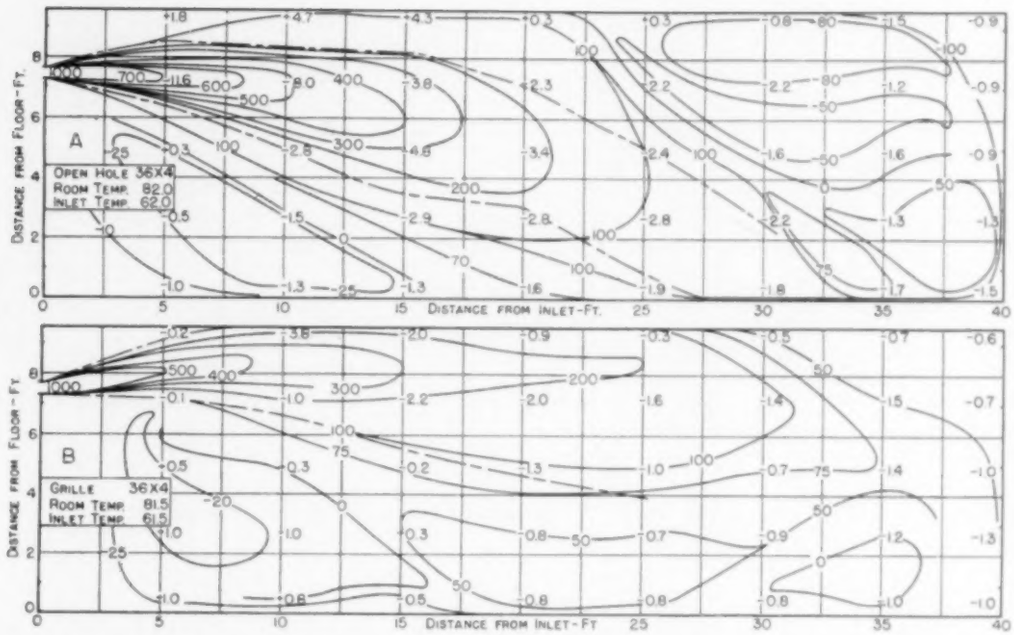

Fig. 8—Air distribution along vertical section through center of room: *A* (*above*) for 36 x 4 open hole; and *B* (*below*) for 36 x 4 grille

able extent. For example, in a high room with the grille mounted near the ceiling, there will be little or no difficulty due to cold air dropping into the occupancy zone, regardless of all other considerations. There remains, however, the problem of making certain that the cooling reaches the far end, although the greater mounting height tends to increase the throw. On the other hand, for room heights usually encountered (except in theaters and auditoriums) the throw is not materially affected by the room height. If the outlet is close to the ceiling, the lessened friction at the upper side of the air stream tends to hold the cold air up. However, if the grille has a considerable upward deflection and is located adjacent to the ceiling, there will be a tendency for the air to rebound from the ceiling and drop into the occupancy zone. It is apparent that the proper type of grille is much affected by the distance between the grille and the ceiling.

As might be expected, a large volume of air has more tendency to drop into the occupied zone than a small volume of air. The larger volume will have a longer throw, all other things being equal. As previously pointed out, a room may be unsatisfactory because too much cold air drops down into the occupancy zone or because it strikes the rear wall and follows that down into the occupancy zone.

As one would also expect, the greater the difference in temperature between that of the room and the entering air, the more tendency there is for the entering air to fall into the occupancy zone.

The effect of the velocity of the entering air is not so simple and is, in fact, very complicated with certain types of grilles. For an open hole, an increase in air velocity will cause the point at which the cool air first drops into the occupancy zone to move toward the rear wall, but may increase the velocity of this objectionable current without appreciably reducing its temperature. The increased entering velocity may raise the velocity down the rear wall to an excessive figure. The effect of velocity upon the flow characteristics from any particular grille can only be determined by a rather elaborate test procedure simulating the conditions of actual use.

Size of the grille is a factor, but, as it is fixed by the volume of air required and the velocity, it does not need to be discussed or considered separately.

Aspect ratio is the width divided by the height of the actual opening inside the supporting frame. It is a factor in distribution, but except when quite large is not as important a factor as appears to be the general opinion. Nevertheless, a high aspect ratio does decrease the throw. If the grille is mounted near the ceiling, the high aspect ratio will increase the rate at which the cool air mixes with the room air and therefore decrease its tendency to fall. If the grille is mounted at a considerable distance below the ceiling (for example, 2 or 3 ft) and is of the upward deflecting type, the effect just referred to will be overcome in many instances by the effect of the high aspect ratio in preventing the air stream from being carried high enough to slide along the ceiling. An example of this combination of circumstances will be seen in the charts of Figs. 7-*B* and 8-*B*.

The practical problem of selecting a suitable grille is often greatly complicated by the width of the room and the necessity of locating the outlets at particular places. For example, a wide room requires either a grille with which the air can be forced to diverge widely from the face of the grille, or a number of outlets must be used. Grilles which give a wide divergence are not suitable for very long throws, and this fact must be kept in mind by the designer. Narrow rooms necessitate the use of a grille which will not give a wide divergence and, generally speaking, such a grille will produce a relatively long throw. A widely divergent grille in a narrow room may result in high velocity down the side wall with consequent discomfort to the occupants near the wall. For architectural reasons it is often necessary to locate the outlet at or near a corner, and for this purpose some sort of outlet must be used which will direct the air away from the adjacent wall. Beams are often encountered. If parallel to the direction of

air flow, they are not ordinarily disturbing, although this is not true if a wide angle grille is necessitated by other considerations. If the beams are at right angles to the only direction from which it is possible to discharge the air, the designer must be sure that the entering air does not strike against the beam, for it will be deflected down into the occupancy zone.

Conclusions

There is a need for an authoritative set of requirements for determining comfortable conditions within a cooled space. Criteria are needed to emphasize that comfort air conditioning specifications should include a relation between the three factors: room temperature, air velocity, and the difference between the room temperature and that of the moving air.

The data published in this paper indicate that for the conditions tested a plain side wall opening is unsatisfactory. More tests and much experience have shown it to be unsatisfactory in most types of rooms. Other tests and experience have likewise demonstrated that nearly all actual installation problems can be solved by use of some type of grille now available.

Acknowledgment

The Research Technical Advisory Committee on Air Distribution is indebted to the Barber-Colman Co. for providing the facilities of their laboratory and the necessary personnel to assist in performing the tests and correlating the data.

Code of Minimum Requirements for Comfort Air Conditioning

Sponsored by Joint Committee of
American Society of Heating and Ventilating Engineers
and the
American Society of Refrigerating Engineers

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SECTION I—PURPOSE, SCOPE AND LIMITATION

THE purpose of this code is to provide a uniform procedure and methods for use in establishing the fundamental basis for the design of comfort air conditioning installations in the United States and Canada. The requirements herein set forth are intended to serve as minimum design requirements, and are not to be construed as limiting good practice nor preventing progress in the art of air conditioning. These requirements are intended to be applied only to systems provided with mechanical circulation of the air.

SECTION II—DEFINITIONS AND ABBREVIATIONS

1. **Comfort Air Conditioning, for the purpose of this code** is defined as the process by which simultaneously the temperature, moisture content, movement and quality of the air in enclosed spaces intended for human occupancy may be **maintained within required limits. If an installation cannot** perform all of these functions, it shall be designated by a **name that describes only the function or functions performed.**

2. **Enclosure,** for the purpose of this code is defined as the space to be served by comfort air conditioning apparatus.

3. **Design Load,** for the purpose of this code is defined as the estimated capacity of the apparatus required to provide and maintain specified inside conditions with specified conditions of temperature and humidity existing outside when the summation of all sources of load that occur coincidentally is a maximum.

4. **Effective Temperature** as defined in the A.S.H.V.E. Guide is an arbitrary index of the degree of warmth or cold felt by the human body in response to temperature, humidity and movement of the air.

5. A.S.H.V.E.—American Society of Heating and Ventilating Engineers.
A.S.R.E.—*American Society of Refrigerating Engineers.*
A.S.A.—*American Standards Association.*

SECTION III—DESIGN TEMPERATURES AND RELATIVE HUMIDITIES

1. **Design Outside Air Temperature and Relative Humidity Conditions** are construed to mean the dry- and wet-bulb temperatures (or relative humidities) specified for design load computations.

2. The design outside dry-bulb temperature in various cities for use in computing the design heating load shall be obtained from the latest edition of The A.S.H.V.E. Guide.

A.S.H.V.E. approved Code by letter ballot vote, January 25, 1938.

3. The design outside dry- and wet-bulb temperatures in various cities for use in computing the design cooling load shall be obtained from the latest edition of The A.S.H.V.E. Guide.

4. **Design Inside Air Temperature and Relative Humidity Conditions** are construed to mean the dry- and wet-bulb temperatures (or relative humidities) specified to be maintained inside of the enclosure at the time of the occurrence of the design load.

5. The design inside dry-bulb temperature for the design heating load shall be assumed at 70 F.

6. The design inside relative humidity for the design heating load shall be assumed as 35 per cent with an assumed outside dry-bulb temperature 30 F.

7. The design inside dry-bulb temperature and relative humidity for the design cooling load, corresponding to the outside dry- and wet-bulb temperature, Section III—paragraph 3, shall be based on an effective temperature selected from the following table, for occupancies in excess of 40 min and for sedentary occupations; between relative humidity limits of 30 and 75 per cent inclusive and the air velocity limits as hereinafter stated in Section VI—paragraph 2.

Design Outside Dry-Bulb Temp. Deg F	Design Inside Air Conditions Effective Temp.
105	75.5
100	75
95	74
90	73
85	72
80	71

SECTION IV—DESIGN LOAD ESTIMATES

1. **Design Heating Load.** The rate at which heat is assumed to be supplied the enclosure shall be the sum of the individual rates as determined by the following items.

2. Heat transfer through the windows, walls, partitions, doors, floors, skylights, ceiling or roof of the enclosure due to the air dry-bulb temperature difference assumed to exist between the air on the opposite sides of the construction considered.

3. Heat to increase the dry-bulb temperature of the outside air and its contained superheated vapor assumed to enter the enclosure by infiltration or purposely introduced for ventilation, whichever is the greater, to the inside design dry-bulb air temperature.

4. Heat to evaporate the necessary water required and superheat the resulting vapor in order to raise the moisture content of the outside air assumed to enter the enclosure by infiltration or purposely introduced for ventilation, to the inside design relative humidity.

5. It shall be clearly stated in both the specifications and proposal if heat credit is to be allowed or permitted for any heat emitting source located within the enclosure.

6. **Design Cooling Load.** The rate which heat is assumed to be extracted from the enclosure to compensate for the sum of the individual rates as determined by the following items.

7. Heat transfer through the windows, walls, partitions, doors, floors, skylights, ceiling or roof of the enclosure due to the air dry-bulb temperature difference assumed to exist between the air on the opposite sides of the construction considered.

8. Sensible heat emission of the occupants within the enclosure. The assumed number of occupants within the enclosure shall be the stated number as designated by the owner as estimated during the time when the sum of the remaining estimated loads is a maximum.

9. Heat emission of electrical, chemical, gas, steam, hot water or other devices, apparatus or lights located within the enclosure.

10. Transfer of solar radiation through the windows, walls, doors, skylights or roof of the enclosure.

11. Heat to be extracted to lower the dry-bulb temperature of the air and its contained super-heated vapor (commonly termed moisture content) assumed to enter the enclosure from the outside to the design inside temperature condition.

12. Heat to be extracted in order to remove the vapor entering the enclosure with the assumed outside entering air including vapor emitted by persons and any other vapor emitting sources located within the enclosure in order to maintain the inside design relative humidity.

SECTION V—SOURCE FOR HEAT TRANSFER, INFILTRATION AND VENTILATION RATES

1. **Heat Transfer Rates for Building Construction.** The unit heat transmission rates for windows, walls, partitions, doors, floors, skylights, ceiling or roof of the enclosure to be employed shall be the rates, for the types of construction involved, obtained from the latest edition of The A.S.H.V.E. Guide, or as may be calculated by employing the conductivity and combined rates of radiation and convection given by this publication.

2. **Heat and Moisture Emitted by Persons.** The estimated rate of heat and moisture emitted by the occupants in the enclosure, for the class of activity involved, shall be obtained from the latest edition of The A.S.H.V.E. Guide.

3. **Heat Emission from Appliances.** The estimated rate of heat emitted by various mechanical, chemical, steam, hot water, gas and electrical appliances located within the enclosure shall be obtained from the latest edition of The A.S.H.V.E. Guide. If any such appliance is located beneath an exhaust hood, served by an exhaust fan, only one-half of the amount stated need be considered. Other known sources of heat within the enclosure such as steam and hot water pipes, etc., shall be included.

4. **Solar Heat Entering Enclosure.** The rate at which solar heat is assumed to enter the enclosure shall be obtained from the latest edition of The A.S.H.V.E. Guide.

5. **Infiltration.** The rates at which air is assumed to enter the enclosure by window and door leakage shall be those for the types of windows and doors involved, obtained from the latest edition of The A.S.H.V.E. Guide. If the rate at which air is specified to be introduced or removed from the enclosure by positive means exceeds the estimated infiltration rate use the greater rate.

6. **Ventilation.** The assumed rate at which outside air is to be positively introduced into the enclosure per occupant, when the contamination of the air within the enclosure results entirely from respiratory processes shall be not less than 10 cfm per stated number of occupants. (See Section IV—paragraph 8.)

The assumed ventilation rates shall be not less than 15 cfm per stated number of occupants in enclosures in which smoking is customarily permitted.

Provision shall be made for air removal from the enclosure either by natural or mechanical means at not less than the assumed ventilation rate.

7. **Air Quality or Purity.** For the purpose of this code the requirements of air quality or purity are assumed to be met if means are provided for the positive introduction of outside air in the amounts set forth in Section V, paragraph 6, and for removal of 95 per cent by count of all dust particles over 10 microns in diameter from all air delivered to the enclosure.

SECTION VI—AIR DISTRIBUTION

1. The quantity, air motion and temperature of the treated air and the method of introducing it to the conditioned space shall be designed to limit the variation in dry-bulb temperature to 3 F or less at a 5 ft level throughout

that portion of a given room that is normally frequented by persons. The temperature shall not be taken at a point close enough to a wall or source of heat to be appreciably affected by it.

2. It is desirable to avoid air velocities exceeding 50 linear feet per minute in the zone between the floor and the 5 ft level, in spaces normally frequented by persons who are not normally moving about.

Exception may be made in the vicinity of a supply or return grille when construction necessity requires the grille to be located below the 5 ft level and in a space normally frequented by occupants.

3. It is desirable, for cooling, that the difference between the temperature of air currents in the space frequented by occupants and the average temperature in such space be not greater than as shown below for the various air current velocities. Velocity and temperature measurements should be made 5 ft above the floor level.

Air Current Velocity Linear Feet per Minute	Temperature Difference Deg F Avg. Room Temp. Minus Air Current Temp.
40 and over	2
Less than 40	3

4. **Requirements for Duct Capacity.** It is desirable that return ducts shall be of sufficient area to convey 100 per cent of the air handled by the fan to permit economical heating and cooling prior to occupancy.

It is desirable, to permit economical operation during mild weather, that the outside air duct be of sufficient area to convey approximately the total quantity of the air handled by the fan and that means be provided for the escape of this air quantity. The outside air duct shall in every case be of sufficient area to permit the minimum ventilation requirements to be met.

SECTION VII—APPARATUS RATING

Fans employed in connection with air conditioning systems shall be rated in accordance with the provision of the A.S.H.V.E. Standard Test Code for Disc and Propeller Fans, Centrifugal Fans and Blowers. Air heaters are to be rated in accordance with the provisions of the A.S.H.V.E. Standard Code for Testing and Rating Steam Unit Heaters. Unit ventilators shall be rated in accordance with the provisions of the A.S.H.V.E. Standard Code for Testing and Rating Steam Unit Ventilators. Air cleaning devices shall be rated in accordance with the provisions of the A.S.H.V.E. Standard Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work. Comfort cooling apparatus and refrigerating machinery are to be rated in accordance with A.S.R.E. Standards, which include Standard Method of Rating and Testing Air Conditioning Equipment and Standard Method of Rating and Testing Mechanical Condensing Units.

SECTION VIII—REQUIREMENTS FOR THE SELECTION OF REFRIGERATING EQUIPMENT

1. City water temperature selected shall not be less than the maximum values obtained from the latest edition of The A.S.H.V.E. Guide.

2. Well water temperature selected shall not be less than the maximum value for the months of July and August for the locality.

3. A wet-bulb temperature shall not be assumed less than the outside design wet-bulb for the rating and selection of cooling towers and evaporative condensers.

4. A dry-bulb temperature shall not be assumed less than the outside design dry-bulb temperature for the rating and selection of air cooled condensers.

SECTION IX—NOISE CONTROL

1. Noise measurements for the enclosure shall be made at a height of 5 ft from the floor. All readings shall be taken 5 ft directly in front of any register face or air conditioning equipment or directly underneath when the register face or air conditioning equipment is located on the ceiling. The normal noise level at each of these locations shall be assumed to be the numerical average of five readings.

2. That portion of the noise resulting from the operation of an air conditioning system shall not be more than as stated in the specifications above the loudness level for the different locations. The enclosure loudness levels during normal occupancy, without the air conditioning system in operation, shall be as stated for average noise levels as given in the latest edition of The A.S.H.V.E. Guide unless otherwise specified.

3. The loudness level above the noise in the enclosure and the noise produced by the air conditioning system shall be expressed in decibels based on the 40 decibel response curve and measure with a sound level meter constructed according to A.S.A. Tentative Standards for Sound Level Meters Z24.3-1936.

SECTION X—GUARANTEES AND EXCEPTIONS

1. **Guarantees** of performance shall be limited to stated performance and capacities that are described and can be measured, and the guarantees shall state the condition under which such capacities or results are to obtain.

2. **Exceptions.** If a proposed installation differs from the provisions in this code, the particular in which the design departs from the code shall be stated in both the specifications and proposal.

Minneapolis Heating Code

The latest revision of the Minneapolis Heating Code governs the construction, installation, alteration, maintenance and repair of all heating and air conditioning plants and equipment in all buildings within the city.

A complete section is included in this code which outlines the minimum requirements for estimating the heat required for warming buildings. Also specific recommendations are given for determining the amount of direct and indirect steam and hot water radiating surface to be installed in complete systems. Other minimum requirements standards deal with selection of boilers, chimney flue areas and pipe sizes for either low pressure steam, vapor and vacuum heating systems. Other sections of this code deal primarily with gravity and mechanical warm air heating plants, as well as ventilation requirements for all types of air conditioning installations. Rules are also given for the installation of oil, gas and stoker automatic equipment.

Studies on Bacterial Control in Air Conditioning

By T. S. Carswell,* H. K. Nason,** and J. D. Fleming*** (*NON-MEMBERS*)
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THE efficiency of the air-washer in removing micro-organisms from circulating air has long been of interest to engineers and sanitarians, but the literature reporting the results of workers in this field is scanty, confused and lacking in conclusive evidence.

The research reported in this paper constitutes an extension of previous studies. In all this work, the aim has been to consider an air-washing unit as a *machine for removal of micro-organisms*. No attempt is made to touch upon other phases of the problem, such as the eventual effect upon the bacterial population of the air-conditioned space, or upon the possible effect upon human health. It is believed that the studies reported in this paper clear up most of the confusion regarding washing of micro-organisms from air, and present a rational picture of the mechanism involved in bacterial removal by air-washing.

In a previous paper[1] Carswell, Doubly and Nason have reviewed the literature relating to bacterial control in air conditioning. This review need not be repeated here except to point out a few particular facts to furnish the necessary background. While it is frequently assumed that water scrubbing of circulating air, as practiced in the average air-washer or humidifier, will eliminate bacteria and molds as well as dusts and pollens, there is actually much evidence to the contrary.

The Whipples[2] are apparently the only other workers on record who reported a reduction in the bacterial content of air after water washing, and their results varied widely from 0 to 100 per cent. Larson[3], on the other hand, in investigating the efficiency of an air-washer in a school building, found that "when using recirculated water the washer supplied bacteria to the air instead of removing them, and even when using new water continuously, it did not show any marked efficiency as a bacteria remover." Clark and Gage[4] showed that the use of air-washer water polluted with *B. coli* resulted in an increase in the number of these organisms in the humidified air of textile mills. Wells[5] showed that in air-washing systems using polluted water, the bacteria of the water were thrown into the air and could be recovered from distant parts of the building. In some cases, most of the bacteria present in humidified air were introduced from the humidifying water.

In 1936, Carswell, Doubly and Nason found that washing effected some reduction in the bacterial content of circulating air and that the addition of a germicide, benzylphenol, to the water gave a much better reduction. It was also found that when sufficient germicide was used in the washing water, the latter was maintained in a practically sterile condition, whereas it rapidly became contaminated when no germicide was added; most, if not all, of the organisms coming from the air being washed.

The investigation herewith reported was undertaken in an attempt to:

1. Obtain fundamental data on the efficiency of the air-washer as an apparatus for bacterial purification of air;
2. Investigate more completely the effect of adding germicide to the washing water; and
3. Formulate a clearer concept of the mechanism of air purification by water washing.

It should be emphasized that this work was directed solely toward obtaining data on the bacterial effectiveness of air-washing equipment and that no conclusions of medical or hygienic significance can or should be inferred.

Apparatus and Methods

Three separate air-conditioning systems were studied which were similar in design. Each system was arranged with an air-washer of the *high efficiency* type in which humidity and temperature control was obtained by washing the air with sprays of chilled water. The systems were installed in a pharmaceutical manufacturing plant, a theater and a general office building, all of which were located in the city of St. Louis.

The study covered summer operation only, and while local atmospheric conditions varied somewhat, cooling and dehumidification were generally being effected. The bacterial and fungal contents of both air and water were studied, and a complete record was kept of all pertinent physical and mechanical data.

The micro-organism content of the air was determined with the aid of an air centrifuge[6] which has been found to give the most accurate results of any method yet devised. Samples of air were drawn through sterile tubes from the air ducts and passed through the instrument.

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**Research Chemist, Monsanto Chemical Co.
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[1]Bacterial Control in Air Conditioning, by T. S. Carswell, J. A. Doubly and H. K. Nason (*Industrial and Engineering Chemistry, 29*, 85, 1937).

[2]Some Preliminary Studies in Air Washing and Its Results, by G. C. Whipple and M. C. Whipple (*American Journal of Public Health, 3*, 1138, 1913).

[3]Tests on the Recirculation of Washed Air, by G. L. Larson (A.S.H.V.E. TRANSACTIONS, Vol 22, 1916, p. 213).

[4]Annual Report, by Clark and Gage, (Massachusetts Department of Health, 1912).

[5]Effect of Polluted Water for Purposes of Humidification, by W. F. Wells and E. C. Riley (Massachusetts Department of Health, Annual Report, 1934, p. 166).

Presented at the 44th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, New York, N. Y., January, 1938.

[6]Apparatus for Study of the Bacterial Behavior of Air, by W. F. Wells (*American Journal of Public Health, 23*, 58, 1933).

IV Report of Sub-Committee on Bacterial Procedures in Air Analysis; A Bacteriological Method of Sanitary Air Analysis, by W. F. Wells, E. B. Phelps and C.-E. A. Winslow (*American Public Health Association* Year Book, 1936-1937, p. 97).

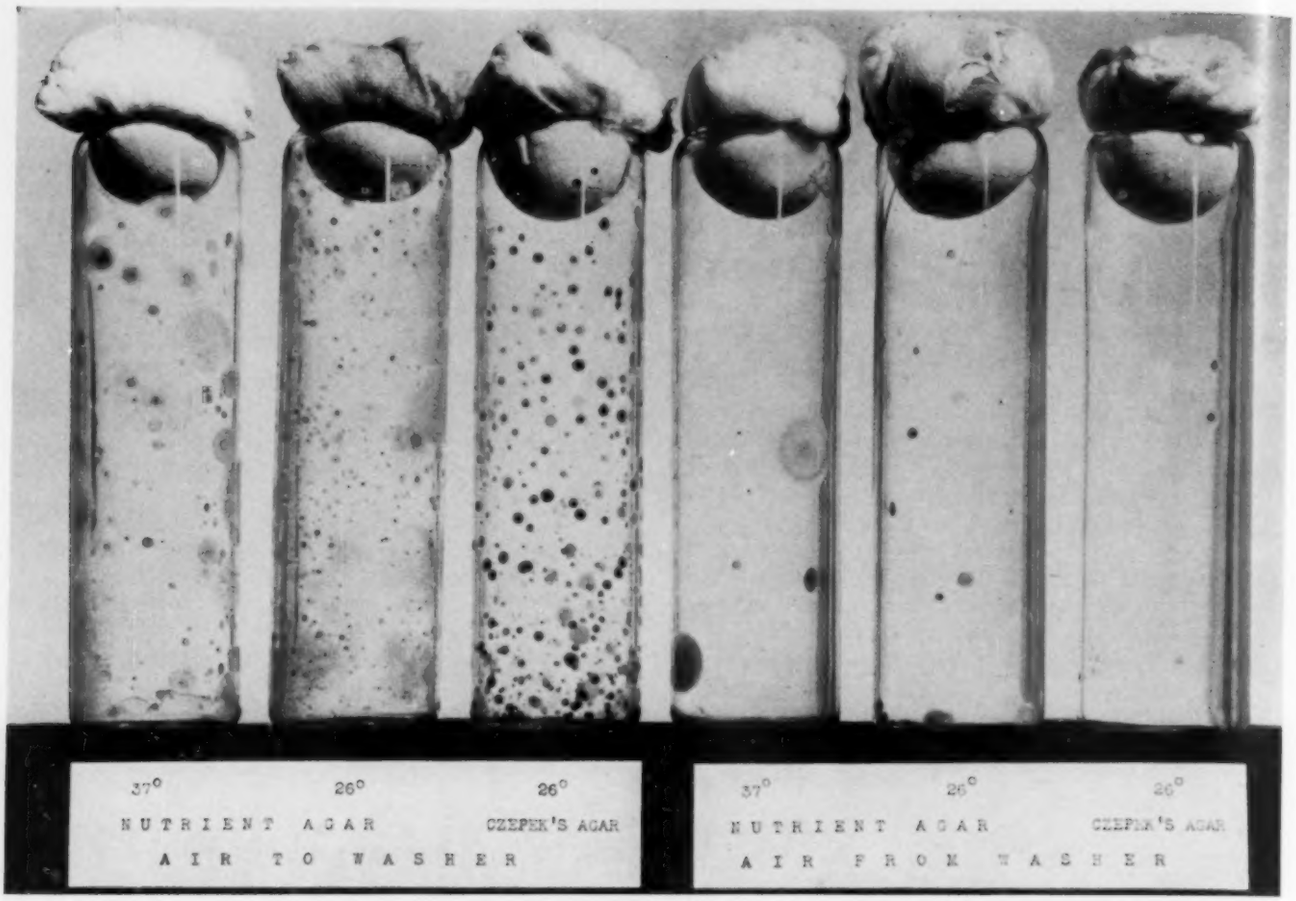

Fig. 1—Bacteria and fungi present in recirculated air, 500 ppm benzylphenol in washing water

The micro-organisms present were deposited by centrifugal force on a layer of culture medium which was supported on the walls of a revolving culture tube. Upon suitable incubation, a visible colony developed on the medium at each point where a living micro-organism had been deposited. The air velocity through the sampling tube was measured continuously by an orifice tube and differential manometer. By sampling for definite times, the sample volumes could be calculated accurately. Corrections for temperature and pressure were made.

Air samples were taken from the main air ducts just prior to entering the washing chamber and several feet beyond the circulating blower on the effluent side of the washer. By comparing the micro-organism content of these samples, the effect of washing could be readily determined. Samples of air-washing water were collected from the circulating pump and micro-organism contents determined by standard methods.[7]

By using various culture media and incubation temperatures, the micro-organisms were roughly separated into the following groups:

1. Bacteria growing at body temperature (possibly parasitic or pathogenic);
2. Bacteria growing below body temperature (probably not parasitic or pathogenic but possibly of industrial importance); and
3. Fungi.

Bacteria were determined by incubation on nutrient agar at 37 C for 48 hours or at 26 C for 72 hours. Fungi were determined by incubation on Czepek's agar at 26 C for 72 hours. Some fungi also appeared on the nutrient agar cultures at both temperatures and were counted separately. The general procedure adopted was to run complete tests for a period of one week under normal operating conditions with no germicide in the wash water. The same tests were then run the following week under conditions as nearly identical as possible except that germicide was maintained in the washing water.

Any germicide used in air conditioning should be highly potent against the micro-organisms involved, noncorrosive to metal used in the equipment, odorless when used in effective concentration, non-volatile, non-toxic to man and to higher animals, economical and safe to use, stable even on prolonged aeration and easily dissolved in water. A commercial mixture of the ortho- and para-benzyl phenols meets these stringent requirements and was used throughout this work. The benzylphenol was dissolved by adding the theoretical equivalent of sodium hydroxide; this was necessary because the free benzylphenol is sparingly soluble in water while the corresponding sodium salt (phenate) is quite soluble.

A benzylphenol concentration of 500 parts per million has been found to be the optimum for insuring sterilization of the water and this concentration was maintained throughout the test period by addition as needed of a stock solution of the chemical. Additional tests have shown that the use of higher concentrations, up to 1,000 ppm, gave no further reduction in bacterial content of either air or water. The amount necessary was determined periodically by chemical analysis, using

[7]Standard Methods of Water Analysis (*American Public Health Association*, 8th Edition, 1936).

a rapid volumetric method, and additions regulated accordingly. By this method, the desired concentration could be closely controlled. Sodium hydroxide solution was also added as needed, to maintain a phenolphthalein alkalinity of about 20 parts per million (expressed as $CaCO_3$), thus insuring against precipitation of the germicide by CO_2 from the air. The alkali also guarded against corrosion by regulation of pH (hydrogen-ion concentration) and hence was of additional value.

The water was changed and the washer cleaned thoroughly before each series of tests. Complete records of temperatures, air velocities and other physical factors were kept for each set of tests.

Table 1—Average 37 C Bacterial and Fungal Relationships (Nutrient Agar)

	Micro-biological Data	Installation: Office Building	Theater	Pharmaceutical Plant
	Bacteria per cc water, no germicide	100	1203	5400
	Bacteria per cc water, 500 ppm benzylphenol	3.2	1.2	12.7
No Germicide	Organisms per 100 cu ft air to washer	269	475	359
No Germicide	Organisms per 100 cu ft air from washer	54	125	277
No Germicide	Reduction, organisms per 100 cu ft washed air	215	350	82
No Germicide	Per cent reduction	80.3	73.7	22.8
500 ppm Benzylphenol	Organisms per 100 cu ft air to washer	388	585	511
500 ppm Benzylphenol	Organisms per 100 cu ft air from washer	76	81	103
500 ppm Benzylphenol	Reduction organisms per 100 cu ft washed air	312	504	408
500 ppm Benzylphenol	Per cent reduction	80.6	84.9	78.1
	Improvement in per cent reduction due to use of germicide	0.3	11.2	55.3

Table 2—Average 26 C Fungal Relationships (Czepek's Medium)

	Micro-biological Data	Installation: Office Building	Theater	Pharmaceutical Plant
No Germicide	Fungi per 100 cu ft air to washer	883	1771	142
No Germicide	Fungi per 100 cu ft air from washer	83	146	64
No Germicide	Reduction, organisms per 100 cu ft washed air	800	1625	78
No Germicide	Per cent reduction	90.8	87.0	57.3
500 ppm Benzylphenol	Fungi per 100 cu ft air to washer	495	2079	303
500 ppm Benzylphenol	Fungi per 100 cu ft air from washer	49	101	51
500 ppm Benzylphenol	Reduction, organisms per 100 cu ft washed air	446	1978	252
500 ppm Benzylphenol	Per cent reduction	90.7	95.0	83.9
	Improvement in per cent reduction due to use of germicide	—0.1	8.0	26.6

Table 3—Average 26 C Bacterial and Fungal Relationships (Nutrient Agar)

	Micro-biological Data	Installation: Office Building	Theater	Pharmaceutical Plant
No Germicide	Organisms per 100 cu ft air to washer	1295	2783	936
No Germicide	Organisms per 100 cu ft air from washer	176	561	342
No Germicide	Reduction, organisms per 100 cu ft washed air	1119	2222	594
No Germicide	Per cent reduction	85.7	80.0	65.6
500 ppm Benzylphenol	Organisms per 100 cu ft air to washer	876	3323	712
500 ppm Benzylphenol	Organisms per 100 cu ft air from washer	159	330	167
500 ppm Benzylphenol	Reduction, organisms per 100 cu ft washed air	717	2996	545
500 ppm Benzylphenol	Per cent reduction	80.8	90.0	74.4
	Improvement in per cent reduction due to use of germicide	—4.9	10.0	8.8

Results Obtained

In all, 526 separate determinations of the micro-organism content of air and 84 determinations of water pollution were made. The data obtained are so extensive that they can only be summarized in this paper.

The results obtained when air was washed with water containing germicide are given in Fig. 1. This photograph shows centrifuge-culture tubes for one set of tests upon the air to and from the washer, which in this case was that installed in the theater. The first tube from the left should be compared with the fourth, the second with the fifth, the third with the sixth, to show the relative microbial content of the air samples before and after washing. The efficient removal of organisms by the air-washer is apparent.

Tables 1, 2 and 3 summarize the micro-biological data developed in these tests. The specific determinations for each test period have been averaged to simplify comparison.

These tables represent data which have been highly condensed from a large number of observations. Table 1 averages both bacterial and fungal relationships as determined by incubation at 37 C on nutrient agar. Table 2 averages fungal relationships only as determined by incubation at 26 C on Czepek's agar. Any bacterial growth shown by the latter plates was disregarded; in fact, only a few bacterial colonies were present, as most bacteria cannot utilize this medium. Table 3 averages both bacterial and fungal colonies incubated at 26 C on nutrient agar. An analysis of fungus and bacterial counts in the case of both Tables 1 and 3 showed that the per cent reduction was approximately the same for either type of growth, and closely equal to the average figures presented in the table for the combined organisms.

A more detailed study of the individual data disclosed the fact that the bacterial efficiency of a washer was nearly constant during the test periods when germicide was used. In the periods when no germicide was used, the efficiency was initially high and was nearly that maintained when germicide was used, but declined steadily and rapidly as the test progressed. The decrease in efficiency was much more marked in those cases where the bacterial pollution of the wash water increased markedly. In one case, when the wash water never became highly polluted, no marked decrease in bacterial efficiency of the washer could be noticed. Figs. 2 and 3 show this effect clearly.

The effect of germicide in the washing water is to render the water sterile and when germicide is used, the bacterial efficiency of the air-washer tends to remain nearly constant. In general, the use of germicide, while maintaining base efficiency does not increase the efficiency beyond that obtained with naturally sterile water. In other words, the germicide acts on the micro-organisms only in the water phase. This is logical when it is considered that air particles are in contact with the germicidal sprays for a fraction of a second only,—too short a time for the germicide to kill. When the organisms are taken into the water phase, however, they may remain in contact with the germicide for several minutes and thus be readily destroyed by the chemical.

The efficiency of the washer will thus be limited only by failure of the organisms to be mechanically taken into the water phase, either because the sprays do not completely cover the cross-sectional area of the washing

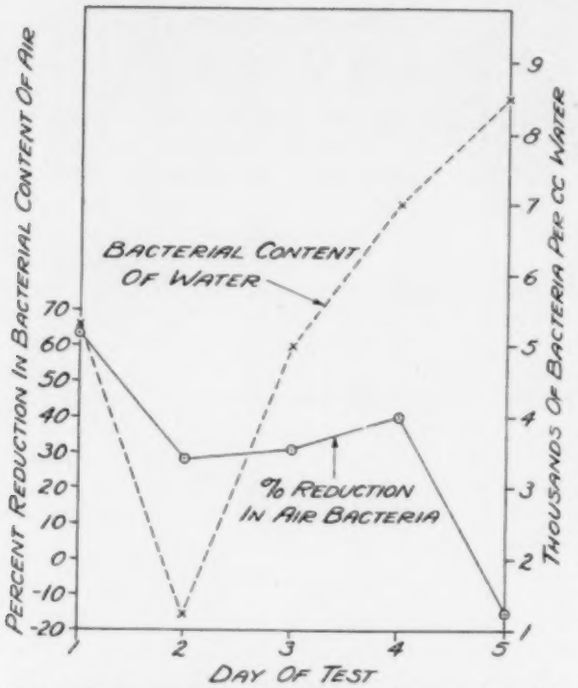
PERCENT REDUCTION IN BACTERIAL CONTENT OF AIR
THOUSANDS OF BACTERIA PER CC WATER
BACTERIAL CONTENT OF WATER
% REDUCTION IN AIR BACTERIA
DAY OF TEST

Fig. 2—Pharmaceutical plant. Relation between bacterial pollution of washing water and bacterial efficiency of air washer (48 hours incubation at 37 C)

chamber, or because part of the air may pass through the sprays without being wetted.

The available evidence also confirms the conclusions of other workers that when highly polluted water is used bacteria may be furnished to the air rather than removed from it. This is shown in Fig. 2 by the negative value for per cent reduction in bacterial content.

That a direct relationship between degree of microbiological pollution of the washing water and the efficiency of the air-washer with respect to removal of micro-organisms does exist is clearly demonstrated by Fig. 4, in which data from all three systems studied are plotted. It will be noted that the increase in bacterial efficiency of the washer when the water was sterilized with germicide varies directly with the degree of contamination of the water before sterilization.

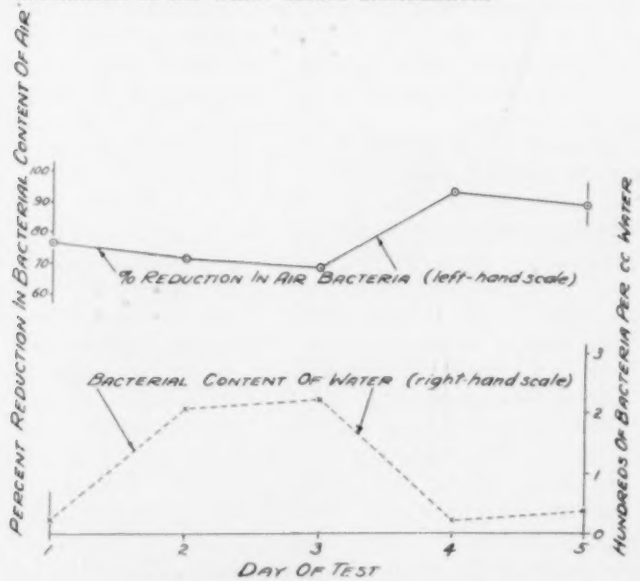
PERCENT REDUCTION IN BACTERIAL CONTENT OF AIR
HUNDREDS OF BACTERIA PER CC WATER
% REDUCTION IN AIR BACTERIA (left-hand scale)
BACTERIAL CONTENT OF WATER (right-hand scale)
DAY OF TEST

Fig. 3—Office building. Relation between bacterial pollution of washing water and bacterial efficiency of air washer (48 hours incubation at 37 C)

Conclusions

Based on the experimental data obtained, a rational picture of the air washing process may be drawn. The conclusions are as follows:

1. When operated on sterile water, an air-washer possesses a certain base efficiency in removing micro-organisms from the air. This efficiency is apparently constant for the particular system (under uniform operating conditions) and is probably a function of the design of the equipment.

2. There is a direct relationship between the degree of microbial contamination of the air-washing water and the efficiency of the air-washer as an apparatus for removing micro-organisms from the air.

3. As the degree of pollution of the washing water increases, the microbial efficiency of the air-washer decreases, and if the pollution becomes sufficiently heavy, the washer efficiency becomes negative. That is, it actually furnishes organisms to the air rather than removing them from it.

4. The addition of sufficient germicide to the washing water to render it sterile insures continuous operation at base efficiency, but does not seem to increase the efficiency beyond this range.

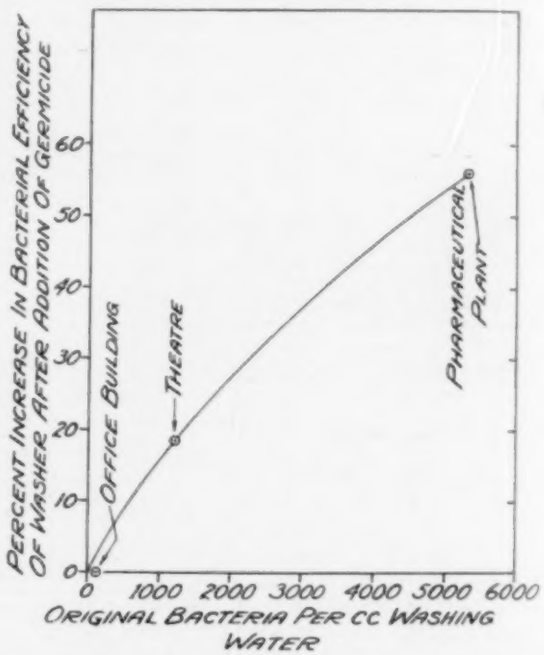
PERCENT INCREASE IN BACTERIAL EFFICIENCY OF WASHER AFTER ADDITION OF GERMICIDE
OFFICE BUILDING
THEATRE
PHARMACEUTICAL PLANT
ORIGINAL BACTERIA PER CC WASHING WATER

Fig. 4—Relation between bacterial pollution of air-washing water and improvement in bacterial efficiency of air washer obtained by sterilization of water with germicide (48 hours incubation at 37 C)

The use of germicide concentrations higher than those required to maintain sterility in the washer water does not increase the efficiency of the air-washer with respect to microbial reduction.

5. Bacteria and fungi are affected similarly by the washing process.

6. An air-washer operated on sterile water is an efficient apparatus for bacterial control of air, and the use of a suitable germicide is a convenient way to insure sterility of the washing water.

Acknowledgments

The authors are indebted to L. L. Lewis and J. H. Bailey of the Carrier Corp. for their aid in securing permission from the owners of the equipment to make these tests. Staffs of the several organizations operating these units rendered valuable assistance and their splendid cooperation is appreciated. R. S. Shumard of the Monsanto Chemical laboratory assisted with some of the bacteriological tests.

Cooling Tower Equipment and Its Relation to Water Conservation

By S. I. Rottmayer* (*MEMBER*), Chicago, Ill.

THE increasing number of cooling-for-comfort installations presents very definite problems to municipalities, especially in metropolitan areas, concerning condenser water supply and disposal. So great has this problem become in certain cities that legislation has been enacted to limit the use of city water for condensers. It follows that water-conserving equipment must be developed for cooling-for-comfort purposes. Such equipment includes means for transferring the heat removed from buildings directly to the outside air using water as the heat-carrying medium.

This paper will deal with a discussion of cooling towers for air conditioning refrigeration, their characteristics and their relation to water conservation, particularly in the Chicago central business district.

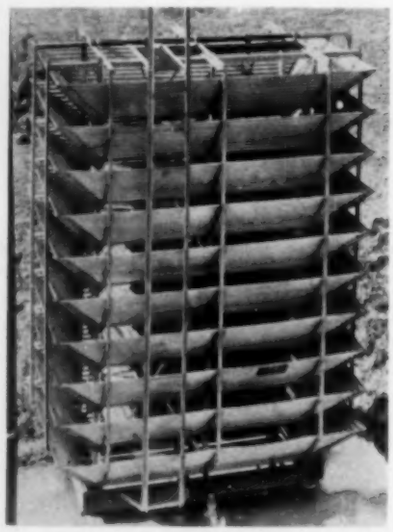

Fig. 1—An atmospheric cooling tower frame; louvers and collecting basin are made of steel

Refrigerating apparatus used in cooling-for-comfort installations performs the function of a heat pump in extracting heat from air at a low temperature level and in discharging it at a higher temperature level. Various refrigerants may be employed by the pump for conveying the heat. In the last stage of the transfer, water commonly is used. Where water costs are low, the easiest course has been to discharge this heat-laden water in the sewer. Water cooling towers are devices in which an intimate contact of water with atmospheric air is achieved, thereby promoting an exchange of heat from the water to the air. The water is cooled in the process, to be used again in the refrigeration cycle.

A successful and economical cooling tower must be capable of breaking up the incoming condenser water so as to present the largest possible surface to the air and at the same time bringing into intimate contact with this water the largest possible quantity of air for the least expenditure of power.

The extent of cooling is limited by the wet-bulb temperature of the air. Hence the efficiency of a device as a water cooler is expressed as a percentage as follows:

$$\text{Per Cent Eff.} = \frac{(\text{hot water temperature} - \text{cooled water temperature}) \times 100}{\text{hot water temperature} - \text{wet-bulb temperature of incoming air}}$$

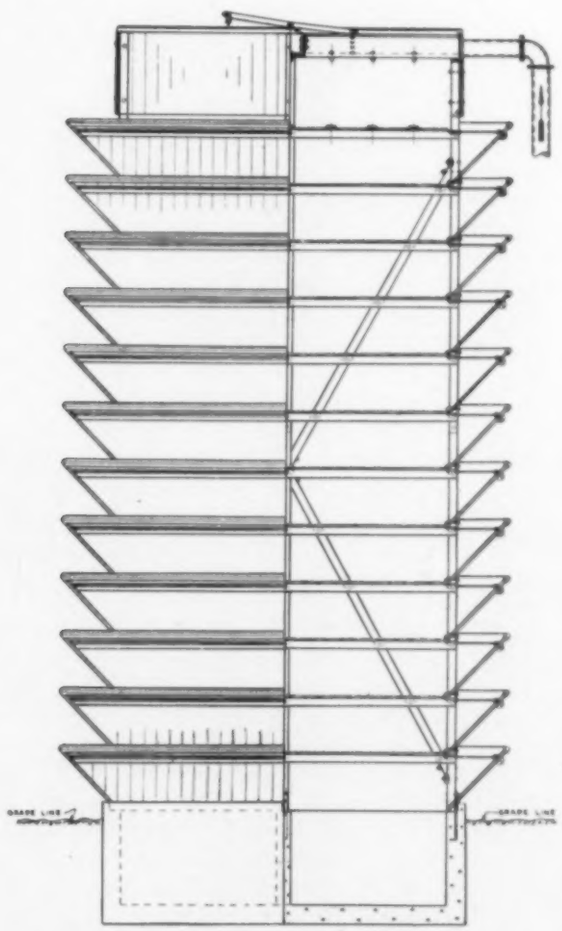

Fig. 2—Half-section and elevation, sectional portable atmospheric cooling tower

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Water cooling towers may be classified as:

1. The atmospheric type which depends on wind motion for operation.
2. The natural draft type, which employs stack or thermal action for producing air movement through the tower.
3. The mechanical draft type, which employs forced or induced draft fans for creating air movement through the tower. The humidifying air washer is a modification of an induced draft cooling tower.
4. The evaporative condenser, which is a combination of a refrigerating condenser and a mechanical draft cooling tower.

Equipment Characteristics

The *atmospheric tower,* called also open natural draft, deck type or wind tower and illustrated in Figs. 1 and 2, is the most common type of tower installed for general refrigerating purposes, considering the number and tonnage of installations. It is constructed of wood or steel framing, adequately braced for wind stress, supporting a filling consisting of a series of horizontal decks or splash bars equally spaced from top to bottom. A system of small louvers or large inclined shields at each deck level

Fig. 3—Atmospheric spray tower

is supported by the outside of the framing. The water enters at the top through troughs, splash heads or spray nozzles and descends from deck to deck to reach a basin at the foot of the tower. The function of the louvers or inclined shields is to prevent loss of water on the leeward side by deflecting the air upward, retaining the water particles blown against them. Vertical, staggered eliminators between the side shields also have been used to reduce further the windage losses or drift of the water.

Increasing the height and length of a tower increases the area of water exposed to air, and increased width increases the length of time of contact of the air with the water.

Since this type of tower depends for operation on natural air movement, it must not be located near obstructions which reduce wind contact and its long side must be at right angles to the prevailing winds. It should be designed to operate at capacity with a wind velocity of not more than 5 mph and with a wet-bulb temperature which is not exceeded during more than 5 to 8 per cent of the season.

Atmospheric spray towers shown in Figs. 3 and 4 are a modification of this type of tower in that the filling or decks are omitted and water is sprayed downward under pressure from nozzles.

Atmospheric towers have advantages of low operating costs and low first cost. Disadvantages are the large size, great area and weight, as well as dependence on variable natural air movement. Spray towers require a higher pumping head and generally are lower in first cost than purely atmospheric towers.

The efficiency of an atmospheric cooling tower will vary with its design and loading usually between limits of 35 and 90 per cent. The usual atmospheric deck type tower has an efficiency of about 67 per cent compared with about 50 per cent for an atmospheric spray tower.

The *natural draft tower* has an air-tight shell enclosing the filler section with air inlets at its base and with its top extending as a chimney above the filler for a considerable distance. An upward flow of air from the bottom through the filler is produced by the difference in weight of air on the inside and outside of the tower, this weight responding to temperature; hence its use is confined largely to application having high entering water temperatures and long cooling ranges. Its performance is not greatly affected by wind velocity.

A *mechanical draft cooling tower* differs from the natural draft type in that the chimney above the enclosing shell is shorter and air is circulated upward by means of fans located either at the bottom or at the top. The water may fall downward trickling over a wood checker work filler or may be sprayed downward inside the shell without any filler. A set of eliminator plates across the top of the tower is provided for removing entrained moisture from the departing air.

A forced draft tower has fans located at the base of the tower as illustrated in Figs. 5, 6 and 7, where they can readily be inspected and serviced. Horizontal vanes may be installed inside the tower to deflect the incoming air evenly across the tower area. An induced draft tower, as indicated in Fig. 8, has fans located above the eliminator plates at the high point of the tower. The fans in such a tower are subjected to the deteriorating effect of the moist air. This tower can be lower in height than the forced draft style for the same cooling results.

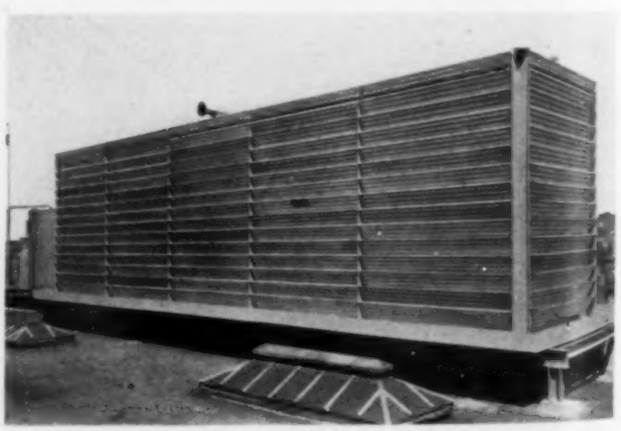


Fig. 4—An atmospheric spray tower of all steel construction

Since the air flow is counter to the water flow, the efficiency of heat transfer of the mechanical draft tower is high. Air velocities over the gross area of the mechanical draft tower generally vary with the design of the tower between limits of 250 and 600 fpm.

Mechanical cooling towers may be located within buildings or may be exposed outside. In many instances they are placed outside as represented in Fig. 9, and are concealed by pleasing architectural treatment. When installed within buildings, mechanical cooling towers may serve as exhaust fans for the ventilating system and may take all or part of the air from inside the building. This air may be cooler and more favorable for service than the outside air.

An ordinary double spray humidifying air washer may be used within a building as a water cooler where considerable area but little head room is available. The high efficiency of counter-flow heat transfer by the vertical mechanical draft tower is not present in this type of apparatus; hence a greater quantity of air must be handled for the same duty.

Disk or propeller type fans generally are used for producing air circulation in mechanical water towers located outside of buildings. Increasing the number of blades of these fans decreases the noise produced. Where particularly quiet operation is essential, housed centrifugal multiblade blowers usually are employed.

Advantages of mechanical draft towers over atmospheric towers are independence of natural wind movement, smaller area and weight. Disadvantages are higher investment, maintenance and operating costs.

The efficiency of a mechanical draft cooling tower ranges between 35 and 90 per cent, depending on its design and loading. The usual design has an efficiency of about 67 per cent.

Fig. 6—Usual style of an all steel forced draft cooling tower

The *evaporative condenser*, as diagrammatically shown in Fig. 10, combines an induced or forced draft cooling tower with a refrigerating condenser. Like an indoor cooling tower, it may take all or part of its air supply from inside the building, and hence may serve also as an exhaust ventilating fan. It is constructed generally with a single bank of spray nozzles wetting the extended coil surface of a refrigerant condenser built into the air passage. A bank of eliminator plates is interposed between the spray chamber and the leaving air stream.

The counterflow principle of heat transfer may be incorporated in this design. For periods of intermediate temperature during the spring and fall this apparatus may be operated without water as an air-cooled condenser.

An evaporative condenser, as its name implies, secures a higher heat transfer rate from the refrigerant to the air than is possible with any arrangement using water alone. This is due to exposure to the cooling air not only of the water but also of the tubes containing the liquid refrigerant. This liquid is many degrees warmer than the air and also is several degrees warmer than the water, and due to a greater temperature difference between the donor of the heat and its recipient, the combined heat transfer

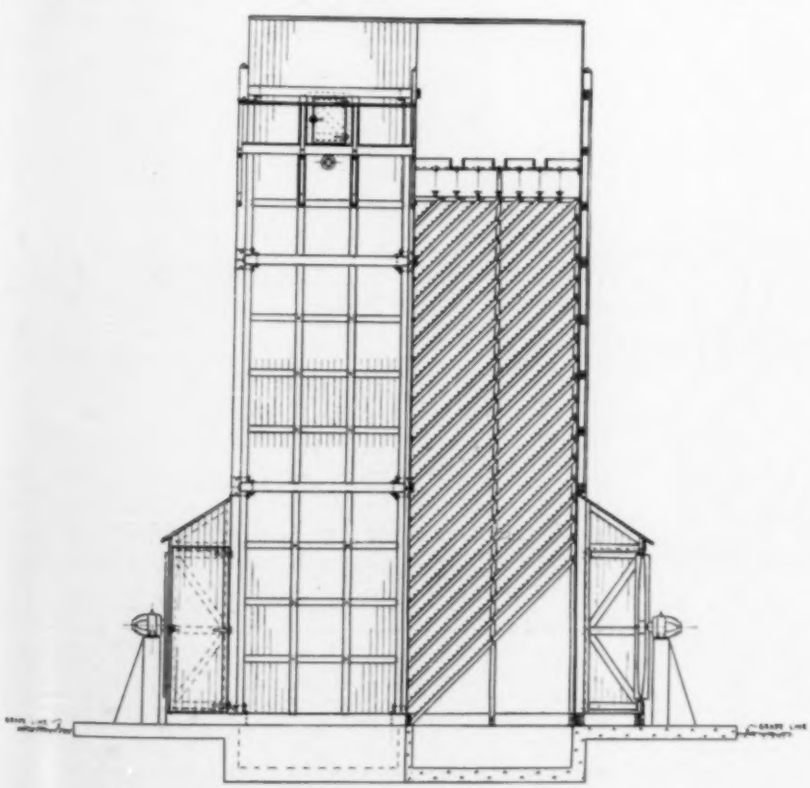


Fig. 5—Half-section and elevation, forced draft cooling tower

rate increases measurably. Much more water is also evaporated per unit of refrigeration, and in addition advantage is taken of the latent heat.

Evaporative condensers usually have been made in comparatively small factory-assembled types, but no particular size-limit needs to exist. They are used with ammonia and the fluorine group of refrigerants having a condensing temperature not warmer than about 100 F.

Make-Up Water Requirements

The quantity of make-up water required by each type of air-contacting equipment is equal to that lost by wind or drift plus that evaporated. The loss in pounds of water through evaporation is equivalent to the quantity of heat to be dissipated, divided by the latent heat of vaporization at the temperature of evaporation, usually slightly above the wet-bulb temperature.

The heat removed per minute in a refrigeration system is 200 Btu per minute per ton plus additions due to leakage, friction, etc. and plus variable additions on account of the ratio of compression, character of the refrigerant, etc.

In the illustrative examples, conditions such as exist in the locality of Chicago will be assumed.

1. Design wet-bulb temperature—75 F.
2. Heat dissipation per ton mechanical compression dichlorodifluoromethane refrigerating system—250 Btu per minute.
3. Heat dissipation per ton, steam jet refrigerating system—737 Btu per minute.
4. Maximum design city water temperature—76 F.

The water loss due to evaporation, being a function of the heat dissipation may be expressed as:

$$\text{Gallons per ton evaporative loss} = \frac{\text{Btu per ton heat dissipation}}{8.33 \times \text{latent heat of vaporization at temperature of evaporation.}}$$

The temperature of evaporation may be taken at 3 F above the wet-bulb temperature.

For the compression system of refrigeration, the evaporative loss will be:

$$\frac{250}{8.33 \times 1094.3} = 0.0275 \text{ gpm per ton}$$

and for the steam jet system:

$$\frac{737}{8.33 \times 1094.3} = 0.08 \text{ gpm per ton}$$

For a dichlorodifluoromethane mechanical compression refrigerating system, with a wet-bulb temperature of 75 F, condenser water temperature range, 85 F to 95 F, the water circulated per ton will be 3 gpm. A cooling range of 7.5 to 10 F is usual with 75 F wet-bulb. The drift loss then is 1 per cent of 3 gpm or 0.03 gpm.

On the basis of 80 F design wet-bulb and a cooling tower of 67 per cent efficiency, then the water circulation required for the steam jet system is:

$$\frac{737}{8.33 \times (95 - 85)} = 8.85 \text{ gpm per ton}$$

The drift loss then will be 0.0885 gpm per ton.

The total evaporation and drift losses per ton of refrigeration, for the conditions stated, are given in Table 1.

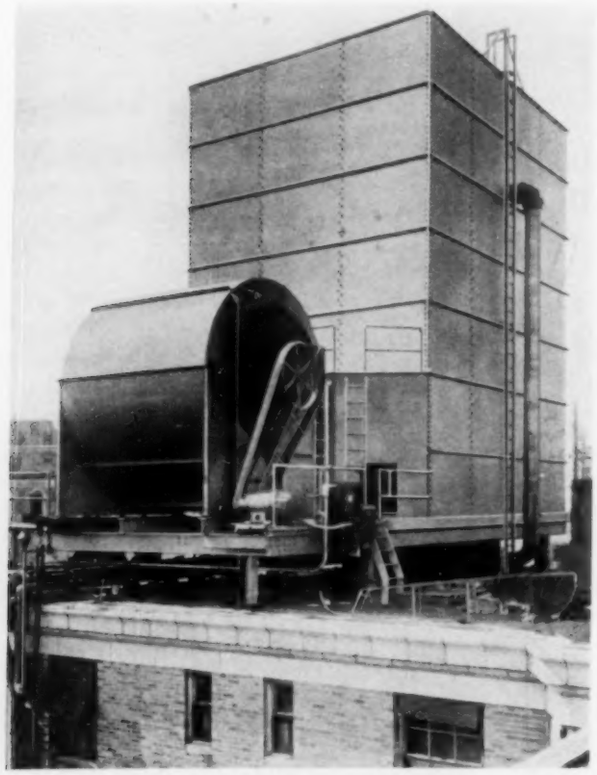


Fig. 7—A forced draft cooling tower installed above hotel guest rooms

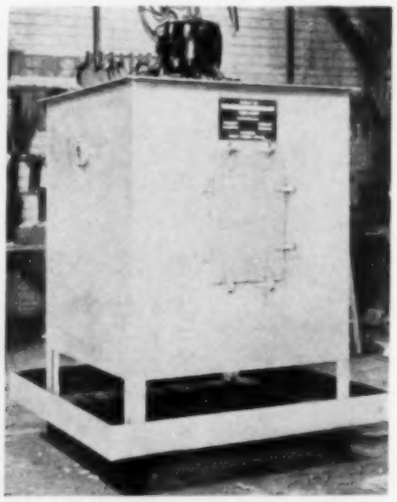


Fig. 8—An indoor induced draft cooling tower

Ordinarily it is considered that in atmospheric cooling about 1 gal of water per 1000 gal of water circulated per degree of cooling range is lost by evaporation and 1 gal of water per 100 gal of water circulated is lost

Table 1—Evaporation and Drift Loss

Refrigerating System	Evaporative Loss, gpm Per Ton	Drift Loss gpm Per Ton	Total Loss gpm Per Ton
Dichlorodifluoromethane Compression	0.0275	0.0300	0.0575
Steam Jet	0.0800	0.0885	0.1685

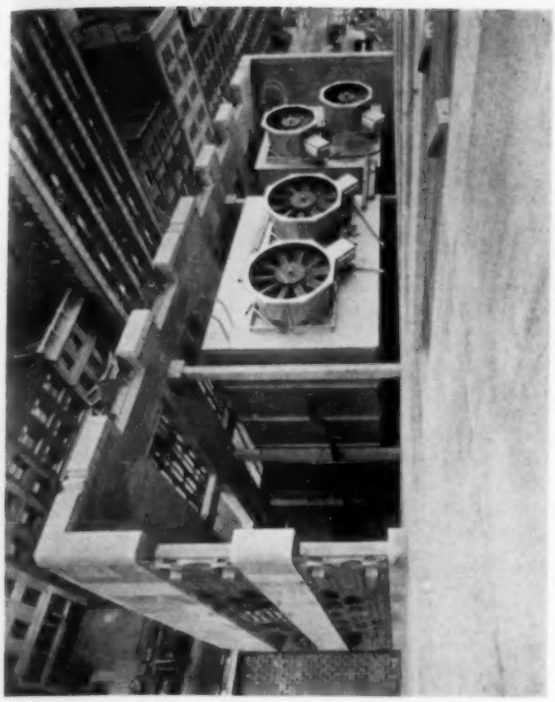


Fig. 9—Induced draft mechanical cooling towers concealed with architectural treatment

by drift. Thus for a plant in which the water enters the cooling tower at 88 F and departs at 80 F and which circulates 500 gpm, the loss will be $\frac{8 \times 500}{1000} = 4$ gpm by evaporation and $\frac{500}{100} = 5$ gpm by drift, or a total of 9 gpm.

If city water is to be used for condensing purposes, with the warmed water wasting to the sewer, the water consumption per ton of refrigeration will be:

Mechanical compression refrigerating system—

$$\frac{250}{8.3 \times (95 - 76)} = 1.58 \text{ gpm per ton.}$$

Steam jet refrigerating system—

$$\frac{737}{8.3 \times (95 - 76)} = 4.65 \text{ gpm per ton.}$$

The saving in water consumption if a cooling tower should be used then will be:

$$\frac{100\,(1.58 - 0.0575)}{1.58} = 96.5 \text{ per cent for the compression system.}$$

and

$$\frac{100\,(4.65 - 0.1685)}{4.65} = 96.9 \text{ per cent for the steam jet system.}$$

A value of 90 to 95 per cent generally is taken as the water conservation effected by the use of cooling tower equipment.

Steam jet refrigeration is especially favorable when cooling tower water is available. As the load varies, following changes in the outdoor condition, the compression ratio between the condenser and the evaporator can be reduced, directly reducing the steam consumption and the demand upon the cooling tower. This wide flexibility of output without corresponding loss in efficiency militates in favor of the steam jet-cooling tower combination, even though a steam jet system requires about three times as much cooling tower condensing water capacity per ton as does a dichlorodifluoromethane compressor system. The latter type when under as little as 25 per cent load, a situation which occurs during many hours, usually requires at least 40 per cent of the full power input. With the steam jet type the efficiency really increases as the load is reduced when cooling towers are employed.

Where water cooling tower equipment is used with steam jet refrigeration it is usual to base the design of the tower on a wet-bulb temperature 3 to 5 F higher than ordinarily is used. This is because steam jets will *flash-back* and fail entirely to function properly if the leaving condenser water exceeds the temperature for which it was designed. The allowance of 3 to 5 F serves as a safety factor for operating insurance.

Carbon dioxide compression refrigeration is not practicable with cooling towers except possibly in exceedingly dry climates, as the critical temperature of this refrigerant is 88 F and the power cost becomes excessive unless the condensing water enters cooler than 80 F. Dichlorodifluoromethane, ammonia and water vapor can be condensed with reasonable success with water which leaves the cooling tower at 85 F.

In the central business district of Chicago, bounded on the North and West by the Chicago River, on the South by Twelfth Street, and on the East by Lake Michigan, there were 16,328 tons of refrigeration installed for air conditioning purposes on October 1, 1937. Of this, only six installations, comprising 620 tons are provided with water cooling tower equipment; the remaining 15,708 tons use city water for condensing and waste the warmed water to the sewer. Of the six installations, one, a 70 ton plant, was installed in 1935 to use city water, a second, a 200 ton plant, was installed in 1936 to use city water. In 1937 forced draft cooling towers were installed for both these plants after it was found necessary during certain periods of the previous years operation to stop the equipment because of inability of the sewers to carry the condenser water. The third and fourth plants of 40 and 10 tons capacity, respectively, were installed with evaporative condensers in 1937. The fifth, a 200 ton plant, was installed in 1937 with a forced draft cooling tower, and the sixth is a low pressure steam jet plant which has been in service for three years with a forced draft cooling tower.

The facilities for supplying water to this central business district are adequate to serve all estimated demands for the next 10 years. The cost of water, 51 cents per 1000 cu ft, is lowest of any city in which there is available information. The sewer facilities in much of the

Table 2—Tabulation of Comparative Costs per Ton of Refrigeration

Type of Water Cooling	First Cost in Dollars	Floor Space in Square Feet	Weight in Pounds	Owning and Operating Costs in Cents per Hour
City Water	2	0	0	0.660
Forced Draft Cooling Tower	40	1	160	0.719
Evaporative Condenser	48	0.65	75	0.782

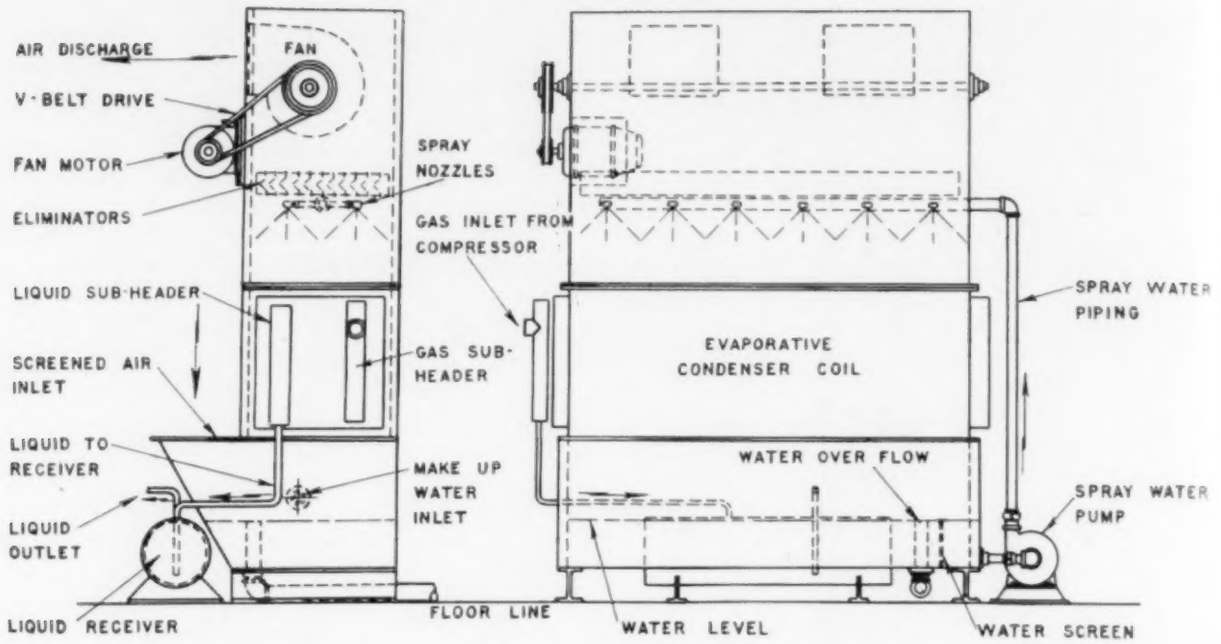


Fig. 10—An evaporative condenser

district, however, are hopelessly inadequate and there are no prospects for their improvement. This situation, therefore, of itself requires that any air conditioning refrigerating plant, if it shall be capable of operating whenever required, shall have its own water cooling tower equipment for dissipating the extracted heat to the atmosphere rather than to the sewer.

The data in Table 2 represent approximate relations per ton of refrigeration for comparative purposes. Exact costs, of course, will depend on the size and specific conditions of the job. Assumptions made for these illustrations in addition to those in the foregoing examples, were: dichlorodifluoromethane compression refrigerating system, all metal construction of cooling towers, 2.5 cents per kilowatthour electricity cost; 51 cents per 1000 cu ft city water cost; 1200 hours operation per season, interest and depreciation computed at 10 per cent for owning cost.

Independent analyses have shown that with present equipment costs and water costs for a load under 600 tons, the cost for the city water-using plant is slightly less than that using cooling tower equipment, and for loads above 600 tons there is a slight advantage to owning and operating cooling tower equipment. An increase in water rates, which might occur should a city filtration plant be installed, and a decrease in cost of cooling tower equipment after its wider usage and development, might easily change the economic balance.

Ultraviolet Lamp in the Hospital

Advantages resulting from sterilized air in hospital operating rooms with the use of the ultraviolet lamp were presented in a paper delivered at a meeting of the American Institute of the City of New York on March 7, 1938, by Dr. Deryl K. Hart, surgeon-in-chief of Duke Hospital, Durham, N. C.

In the early days of the hospital, despite every precaution, an occasional wound infection occurred. After much preliminary work and attempting to eliminate these infections, it was found that the Staphylococcus aureus, which caused over 90 per cent of the infections, was regularly present in the air.

It was demonstrated that the outside air was practically free of pathogenic bacteria; that the air after being washed and filtered was free of all organisms, but that the air leaving the operating room was highly contaminated. After painting and scrubbing the operating rooms to get them free of organisms, it was determined that they were rapidly recontaminated by the entrance of human beings.

Turning to radiation of known bactericidal properties in an attempt to sterilize the air, it was shown that various types of bacteria could be killed within a very short period of time at a distance of 5 ft from the source of radiation. It was then shown that the bacterial content of the air at the operative site could then be reduced to a very low level by this radiation.

Petri dishes of blood agar were sprayed with various organisms and exposed to the radiation of the intensity used for operating, and it was found that these could be killed at the operative site within from 1 to 3 min of exposure. Some of the fungi were more resistant but most of these were killed within from 5 to 20 min exposure. Various hospitals throughout the country cooperated by exposing plates in their operating rooms with similar results. These studies also showed that air conditioned operating rooms had a much lower bacterial content than non-air conditioned rooms.

Odor Control in Animal Laboratories

By F. H. Munkelt* (*MEMBER*), New York, N. Y.

A SERIES of tests was made recently in an animal laboratory for the purpose of determining the effect of various air quantities in reducing the odor intensity to a satisfactory level. The tests were made with various proportions of outside and recirculated air. Except in one of the tests, all of the recirculated air was passed through adsorbers of activated coconut shell carbon. The data secured were to be used in connection with the design of a modern animal laboratory.

Description of Test Laboratory

A room used for storing and breeding rats and guinea pigs was selected and partitioned to form a room 19 ft 9 in. wide by 25 ft 2 in. long by 9 ft 6 in. high, with a gross content of 4722 cu ft. The floor and one outside wall were concrete, while two of the walls were old wooden partitions. The fourth wall, or new partition, was of composition board. The ceiling was the wooden floor and exposed beams of the second story.

The animal contents of the room consisted of the following:

500 adult rats, average weight	.....	275 grams each
600 young rats, average weight	.....	40 grams each
48 guinea pigs, average weight	.....	325 grams each
Total Weight	.....	390 lb.

The animals were caged in standard rows and tiers. In some cages they were bedded in shavings; in others, they walked on wire mesh under which were removable trays for cleaning purposes. Odors from other sources than from the animals were negligible, as there were never more than two observers present in the room. It is possible, however, that the two, old, wooden partitions and the ceiling, which had been in place for a long period, absorbed and regenerated animal odors to some extent. This condition would not apply to a modern room faced with glazed tile or glazed brick.

Arrangement of Test Equipment

The apparatus was arranged as illustrated in Fig. 1. The adsorber A is rated at 1000 cfm and is the standard type of vertical adsorber commonly applied to air conditioning and ventilating systems. Air was admitted from the room to the top of the adsorber casing, was passed through a dust filter, and then through odor adsorbers to the fan. A 6 in. outside air duct B, with measuring tube F, was led to the adsorber casing below the odor adsorbers to permit mixing of outside air with air which had been passed through the odor adsorbers. A belt driven fan D distributed the mixture uniformly to the opposite side of the room through a simple duct system C with seven openings G, each 7.5 in. in diameter.

*Vice-Pres., Consolidated Air Conditioning Div., W. B. Connor Engrg. Corp.

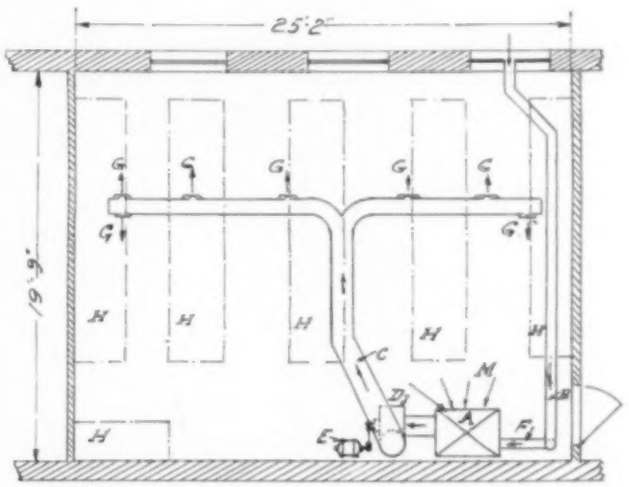


Fig. 1—Plan of experimental room

The odor adsorber consisted of 40 standard canisters screwed vertically to a flat, horizontal manifold plate, which, in turn, was mounted in a vertical casing as previously described. Each canister was 4 in. in diameter by 10.5 in. high and consisted essentially of two concentric, cylindrical, perforated shells containing in the space between them a bed of granular, coconut shell activated carbon. The air stream passed through the perforated plates and the activated carbon and then downward through the manifold plate. The quantity of air passed through each canister varied in the several tests from 16.25 cfm to 23.1 cfm, and the resistance from 0.07 in. to 0.13 in. water gage.

Variations in air quantities were obtained by setting dampers on the discharge duct and on the outside air duct and by pulley changes on the fan. Before assembling, the adsorber and the measuring tubes were carefully calibrated so that the air quantities could be read on draft gages.

The odor was typical of that generally found in animal houses, monkey houses, zoos, etc. and consisted of a mixture of organic acids and other compounds arising from the metabolisms of the animals and given off by the excreta. Ammonia, arising principally from the decomposition of the excreta, was also present, but it could not be detected easily by the sense of smell when other animal odors were present.

Test Procedure

Five odor levels were chosen for recording the results, namely, *very strong, strong, moderate, definite* and *threshold*. Observations were taken before the start of each test and at 30-min intervals until a constant odor level was reached. Two or three observers were used in each test. The observers entered the room from outdoors and recorded their observations separately.

Table 1—Odor Control Factors for Various Tests

Factors	7-15-37 Test 1	7-15-37 Test 2	7-16-37 Test 3	7-19-37 Test 4	7-24-37 Test 5
Room air recirculated through carbon, cfm	800	650	500	900	925
Air admitted from outside, cfm	210	150	80	230	230
Total clean air circulated, cfm	1,010	800	580	1,130	1,155
Room volume net, cu ft	4,700	4,700	4,700	4,700	4,700
Dilution rate, air changes per hour	12.9	10.2	7.4	14.4	14.7
Room air through bypass,[a] cfm			170		
Total fan delivery, cfm	1,010	800	750	1,130	1,155

[a]Door on fan suction to admit room air.

The curves showing the odor intensity, on Figs. 2 to 6 inclusive, are the average of the observations. Separate observations were made at the outlets G (Fig. 1) and a separate curve recorded. The odor control factors for each of the tests are recorded in Table 1.

In Tests 1, 2, 4 and 5, the outside air admitted was approximately 20 per cent of the total air circulated. Test 4 and Test 5 do not differ materially in the air quantities chosen. However, Test 4 was completed just before a regular cleaning of the laboratory was begun and, accordingly, it was decided to run another test, No. 5, at a time when the laboratory was relatively clean. The cleaning dates were July 5th and July 19th. Test 5 was made on the morning of July 24th, less than five days after the laboratory had been thoroughly cleaned.

In the case of Test 3, a total air circulation of 750 cfm was used although only 580 cfm of this amount was either outside air or passed through the odor adsorbers. Test 2, with smaller air quantities than Test 1, had shown a less favorable odor level than Test 1. In Test 3, the air quantities were still further reduced and the results confirmed the trend in Test 2.

In Test 1, Fig. 2, 800 cfm were recirculated and 210 cfm were drawn from outside air, resulting in a total of 1010 cfm, or 12.8 air changes per hour. The curve shows a diminishing animal odor intensity at each successive 30-min period until it reached a satisfactory level, slightly over *threshold* and well under *definite* at the end of 120 min.

Test 2, Fig. 3, was conducted in the afternoon of the same day as Test 1 and was begun one hour after the completion of Test 1. The air quantities were reduced to 650 cfm of recirculation and 150 cfm of outside air, with a total of 800 cfm, or 10.2 air changes per hour.

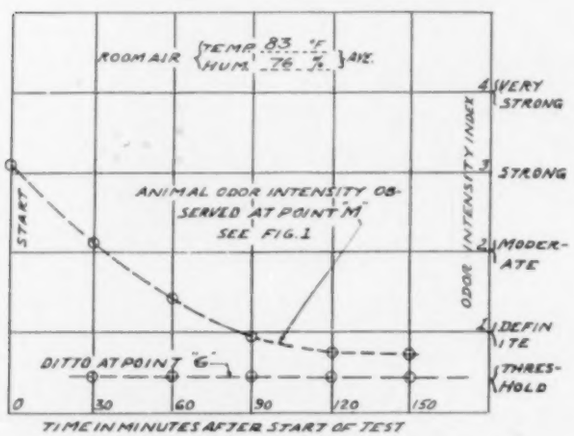

ROOM AIR {TEMP. 83 °F HUM. 76 %} AVE.
ANIMAL ODOR INTENSITY OBSERVED AT POINT "M" SEE FIG. 1
DITTO AT POINT "G"
START
4 VERY STRONG
3 STRONG
2 MODERATE
1 DEFINITE
THRESHOLD
ODOR INTENSITY INDEX
0 30 60 90 120 150
TIME IN MINUTES AFTER START OF TEST

Fig. 2—Odor observations for Test 1 started at 9:20 a. m. on July 15, 1937

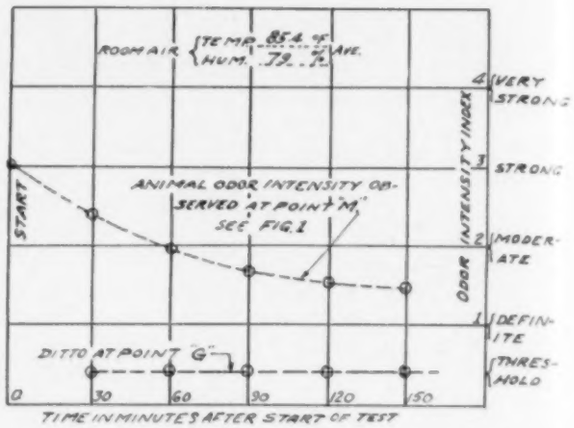

ROOM AIR {TEMP. 85.4 °F HUM. 79 %} AVE.
ANIMAL ODOR INTENSITY OBSERVED AT POINT "M" SEE FIG. 1
DITTO AT POINT "G"
START
4 VERY STRONG
3 STRONG
2 MODERATE
1 DEFINITE
THRESHOLD
ODOR INTENSITY INDEX
0 30 60 90 120 150
TIME IN MINUTES AFTER START OF TEST

Fig. 3—Odor observations for Test 2 started at 1:30 p. m. on July 15, 1937

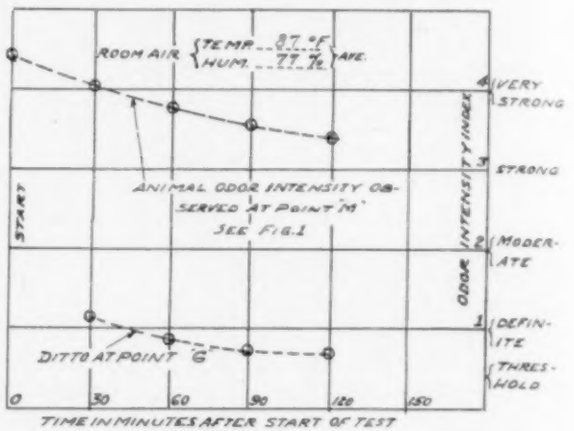

ROOM AIR {TEMP. 87 °F HUM. 77 %} AVE.
ANIMAL ODOR INTENSITY OBSERVED AT POINT "M" SEE FIG. 1
DITTO AT POINT "G"
START
4 VERY STRONG
3 STRONG
2 MODERATE
1 DEFINITE
THRESHOLD
ODOR INTENSITY INDEX
0 30 60 90 120 150
TIME IN MINUTES AFTER START OF TEST

Fig. 4—Odor observations for Test 3 started at 10:00 a. m. on July 16, 1937

It should be noted that the odor level had increased in the one hour period between tests to *strong* and that, after the test was begun, the odor level was found to be reduced each 30 min until, at the end of 150 min, it was approximately level at a point considerably higher than in Test 1, namely, about half-way between *definite* and *moderate*.

Test 3, Fig. 4, was run on the following morning after the room had been closed with virtually no ventilation during the night. The air quantities were reduced to 500 cfm recirculation, 80 cfm outside air, a total of 580 cfm, or 7.4 air changes per hour, calculated from the amount of outside air plus the amount of air passed through the odor adsorbers. The fan delivered 750 cfm, of which 170 cfm were drawn into the fan suction through an open door and were not treated in any way. The temperature, as well as the humidity, was very high and, undoubtedly, contributed to more rapid odor generation. Another reason for high odor generation was the fact that the test was run only a few days before the regular periodic cleaning of all of the cages and the room. It should be noted that the odor intensity at the start was observed to be higher than *very strong*. As the test proceeded, the odor level dropped but only to a point still considerably above *strong*.

Test 4, Fig. 5, was conducted on the morning of the day of the regular periodic cleaning of the room. The

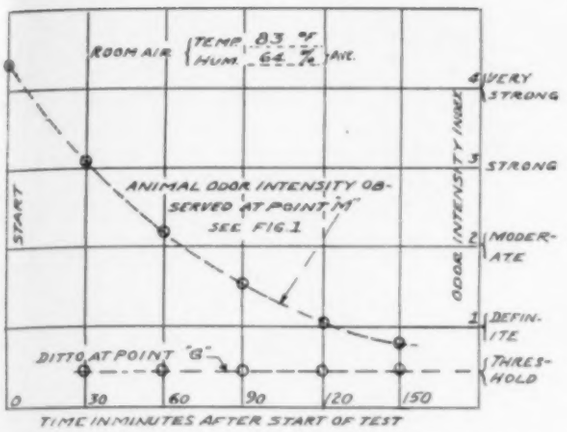


Fig. 5—Odor observations for Test 4 started at 10:00 a. m. on July 19, 1937

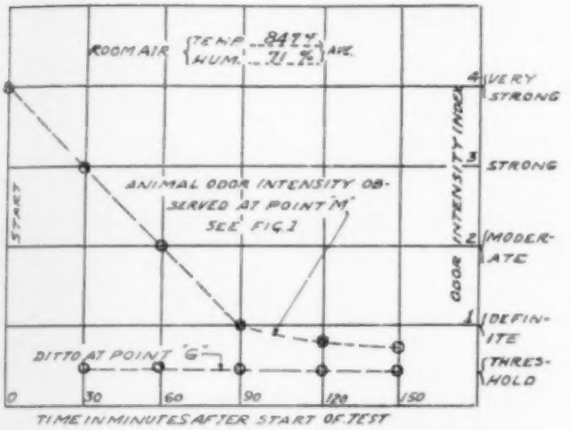


Fig. 6—Odor observations for Test 5 started at 10:10 a. m. on July 24, 1937

air quantities selected were 900 cfm recirculation, 230 cfm outside, or a total of 1130 cfm, or 14.4 air changes per hour. The odor was reported above *very strong* at the beginning, but it was reduced rapidly until it reached a level near *threshold* at the end of 180 min. This is the same level reached by Test 1 but required a somewhat longer period in spite of a slightly increased air change.

Test 5, Fig. 6, was run on the fifth day after the cages and room were thoroughly cleaned. The air quantities were approximately the same as in Test 4, namely, 925 cfm recirculated air and 230 cfm outside air, or a total of 1155 cfm, or 14.7 air changes per hour. Due, no doubt, to the cleaner condition of the cages, a much steeper curve was observed, reaching nearly to its final level after about 90 min.

On each of Figs. 2 to 6 inclusive, an additional graph is shown near the bottom reporting observations at points G which are the delivery openings in the ducts. In every case, except Test 3, Fig. 4, this graph is plotted below the *threshold* level as no animal odor was noted at the duct delivery points. In Test 3, 170 cfm, or 23 per cent of the total air delivery was untreated and carried the same odor content as the room air. It would be expected, therefore, that the odor noted at points G would be considerably below the odor intensity in the room, but, nevertheless, quite definite. It should be noted that the observers placed the odor intensity at points G, Test 3, as slightly below *definite* and above *threshold*.

The ammonia odor was considered separately. Ammonia is not readily adsorbed by coconut shell activated carbon and in the type of test apparatus used the adsorption was too small to warrant consideration. In all of the tests it was observed that, as the animal odors were reduced, a faint ammonia odor became perceptible. Generally, the odor increased for a period and then diminished again until it became constant. The ammonia odor was noted more easily at points G, the outlets of the ducts, than in the room where the presence of animal odors tended to mask it.

The ammonia odor concentration in the room was always at a sufficiently low level in Tests 1, 4 and 5, which, it will be noted, provided outside air changes of 2.7, 2.9 and 2.9 respectively. Undoubtedly, the displacement of room air by outside air assisted in the reduction of the ammonia odor level. This conclusion is further confirmed by the observations of ammonia odor in Test 3, which provides only one outside air change per hour. In this test the ammonia odor in the room became quite strong after the test had continued for one hour and it did not diminish thereafter. At the duct outlets G, the ammonia odor was found to be *very strong* at the end of one hour. It was, in fact, strong enough to be irritating to the nasal passages of some of the observers.

General Summary

These tests demonstrate the practicability of reducing the odor in an animal laboratory to a satisfactory level by the use of recirculated air from which the odors are removed by an odor adsorption process. The odor level obtained will depend upon the quantity of air which is circulated per minute, and to a lesser extent upon the degree of cleanliness maintained in the cages. A portion of the air should be brought in from the outside for ventilation as well as for ammonia displacement.

An examination of the test diagrams indicates that a satisfactory odor level in animal laboratories occupied by small animals, such as rats and guinea pigs, can be reached by providing a circulation of 0.75 cfm per animal, or of 2.25 cfm per pound of animal. The latter value, however, is probably higher than necessary. In the tests about half of the animals were designated as young rats weighing about ⅓ that of an adult rat. Because of the necessity of bedding the young rats in shavings, their odor generation was, without doubt, relatively high as compared to adult rats. A value of about 1.5 cfm per pound of animal should be found entirely satisfactory. These conclusions are based on a room volume of 4 cu ft per animal, and an air change of about 11 per hour and a reduction of the odor level to *definite* or lower.

Because of the presence of small quantities of ammonia, an outside air quantity of at least 20 per cent should be provided, which corresponds to a value not less than 0.15 cfm per animal, or not less than 2.5 changes per hour. These tests indicate that outside air in these quantities displaces ammonia at a sufficient rate to maintain satisfactory conditions.

Correlating Thermal Research

As a part of the efforts of the A. S. H. V. E. Research Laboratory to correlate research in thermal engineering carried on by the many institutions engaged in such work, and to disseminate the published results of such studies together with other reports of progress in the field, and in order to make this information available to the membership of the Society, there will be published monthly on this page a limited number of brief abstracts of articles which it is believed will be of interest to all concerned. For more complete lists address the Librarian, A. S. H. V. E. Research Laboratory, U. S. Bureau of Mines Experiment Station, Pittsburgh, Pa.

F. C. Houghten, *Director.*

W. L. Fleisher, *Chairman,*
COMMITTEE ON RESEARCH.

• Does Preheated Air Decrease Flue Gas Temperatures? by J. R. Darnell. *Power Plant Engineering,* Vol. XLII, No. 2, Feb. 1938, pp. 142-143. Author discusses his reasons for believing that preheated air decreases flue gas temperatures. References are given from literature, showing that some tests have been made which seem to uphold his contention.

• Protective Coatings for Underground Piping Systems. U. S. Dept. of Commerce, National Bureau of Standards, Washington, D. C. Letter Circular LC-510, Jan. 3, 1938, 26 pages. Abstracts and summaries of 29 articles and reports, relating to protective coatings for underground piping systems, show recent investigations and developments in this field. Title and author of article are given, also publication data in one or more magazines showing where the article may be obtained.

• Symposium on Corrosion Testing Procedures. *American Society of Testing Materials,* 131 pages, 7 technical papers. Price $1.25. Papers include: principles of corrosion testing; atmospheric and salt spray testing; electrical resistance method of determining corrosion rates; corrosion testing methods for copper alloys; standardizing liquid corrosion tests; and soil corrosion testing.

• Air Conditioning of Railroad Passenger Cars, L. W. Wallace and G. C. Early. *American Society of Mechanical Engineers, Transactions,* 59, pp. 733-740, Nov. 1937. Brief sketch of development of railroad air conditioning and methods of refrigeration used for cooling passenger cars. Results of passenger-car air conditioning study in 1936. Tables and curves are given showing the total cost of the different methods of air conditioning for a variety of operating conditions.

• Power for Air Conditioning, by A. D. Marston. *Power Plant Engineering,* Feb. 1938, Vol. XLII, No. 2, pp. 125-126. Discussion of method of estimating hours of operation per month for an air conditioning system. Data are given for summer cooling for the Kansas City district, with an example using data for a restaurant with 20 hp installation.

• Air Conditioning with Ice, by C. D. Robinson. *Ice and Refrigeration,* Vol. XCIV, No. 2, Feb. 1938, pp. 105-106. Results of an experiment in the use of ice as cooling medium for summer air conditioning of a four story office building and an eight room residence in Omaha. Cost of installation of air conditioning system and yearly operating costs.

• Refrigeration Storage Applied to Air Conditioning. *Refrigerating Engineering,* Vol. 35, Nos. 2, and 3, Feb. 1938, pp. 92-96 and March 1938, pp. 178-182. Refrigeration storage can modify the characteristics of air conditioning load so as to benefit both the customer and the utility company, and lower power costs for an air conditioning service. Discussion of economic background of refrigeration storage or hold-over. Data on cost of power supply for commercial air conditioning loads. Description of methods for storing refrigeration and construction details for some typical installations. Determination of economic amount of storage desirable. Details of some refrigeration-storage systems now in operation.

• Extended Surface Cooling Units, by R. H. Swart. *Refrigerating Engineering,* Feb. 1938, Vol. 35, No. 2, pp. 107-115. Mathematical discussion of design calculations of extended surface cooling units. Study of type of refrigerants, air characteristics desired, size and material of tubing and material and design of fins.

• Strength, Water Absorption, and Resistance to Freezing and Thawing of Sand-lime Bricks, by John W. McBurney and A. R. Eberle. U. S. Dept. of Commerce, National Bureau of Standards, Research Paper, RP1065, Vol. 20, Jan. 1938, pp. 67-76. Report of tests to determine the possibility of providing measures of the durability of sand-lime brick, based upon correlation between physical properties and resistance to freezing and thawing. Data on bricks from 11 plants from principal brick producing centers of U. S., methods of tests and conclusions.

• Humidity Measurement, by Arthur W. Ewell. *Refrigerating Engineering,* Vol. 35, No. 3, March 1938, pp. 176-177. Discussion of measurement of humidity in air at low temperature. U. S. Weather Bureau tables of relative humidity below 32 F are much higher than similar English and German tables. Data on relationship of wet- and dry-bulb temperatures below freezing and per cent of relative humidity in air.

• Savings in Fuel Consumption. *Domestic Engineering,* Vol. 151, No. 3, Mar. 1938, pp. 54-55. Discussion of reduction of fuel consumption by use of reduced night temperatures due to thermostatic control.

• Smartly Run School Plant Saves $6600 a Year on Slim Budget. *Power,* Jan. 1938, Vol. 82, No. 1, pp. 40-41. Discussion of survey made at Fullerton Union High School which saved the school $6600 per year. Description of several minor changes made to improve operating conditions.

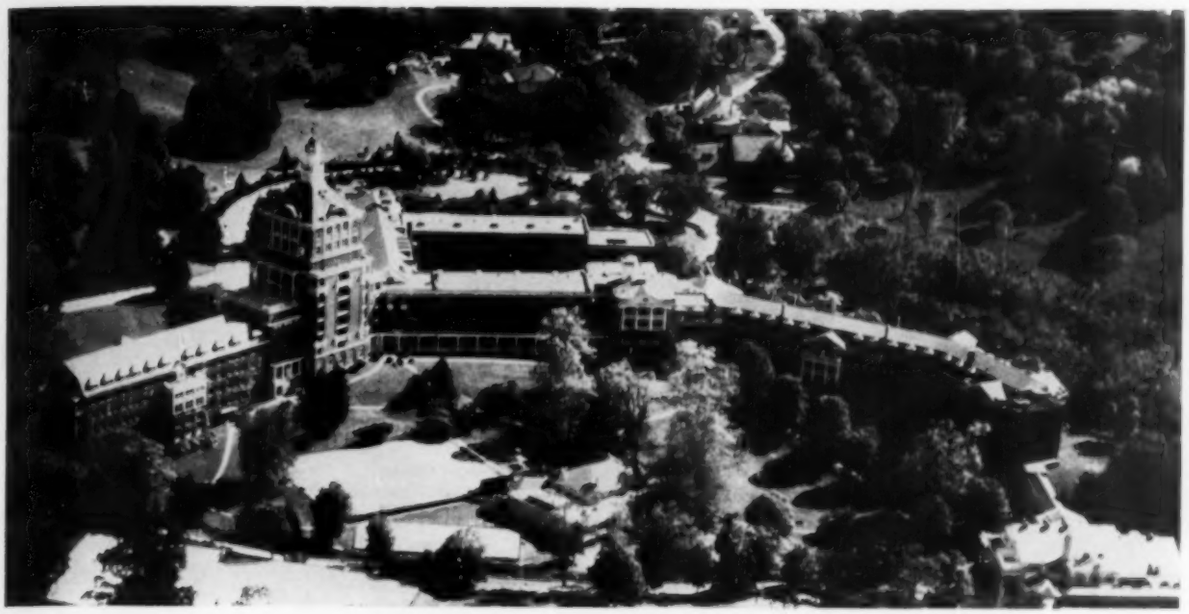
• First City of Steel Houses Planned for U. S. Steel Corporation. *The Iron Age,* Feb. 17, 1938, pp. 80-82. Description of small homes to be built for employees of Carnegie-Illinois Steel Co. at new Irvin, Pa. works. Also description of five-unit all-steel prefabricated farm buildings, including house, barn, poultry house and smoke shed, which is to be available unassembled at $2000. Farm buildings are part of Federal F. S. A.'s program for developing adequate dwellings for low-income farm families. Buildings will be fireproof, lightning-proof, vermin-proof and termite-proof.

• Fundamentals of Fuel Oil Combustion, by E. G. Roberts. *Power Plant Engineering,* Vol. XLII, No. 2, Feb. 1938, pp. 111-114. Theory of burning fuel oil in the combustion zone and some facts and conclusions developed therefrom. An outline of the general process of burning as a guide to improvement of operation in a power plant furnace. Fundamental requirements for efficient operation listed for oil-fired furnaces.

• Harmful Industrial Dusts, R. R. Sayers, U. S. Public Health Service. *Public Health Reports,* Vol. 53, No. 6, Feb. 11, 1938, pp. 217-228. Description and discussion of various kinds of dust. Organic dusts discussed under headings of non-living dusts, toxic and irritant; living dusts such as bacteria and fungi; also inorganic dusts. Methods of dust control from point of view of the engineer and the medical man.

• The Fabric of Modern Buildings, by E. G. Warland. Published by Sir Isaac Pitman and Sons, Limited, London. Price 20s net. Indication of various problems of framed structures of today, especially those of mechanical equipment and insulation against noise. Discussion of materials used from foundation to the roof of several new steel frame buildings. Only a little consideration of reinforced concrete. Many isometric illustrations.

• Ventilatoren, by B. Eck. Published by Julius Springer, Berlin. 197 pages. Price 13.50 rm. Treatise on design, construction and operating characteristics of centrifugal and screw blowers.



Air View of The Homestead, Hot Springs, Va.

Hot Springs for the Summer Meeting

The Council of the Society has selected a delightful spot in the heart of the Allegheny Mountains, The Homestead at Hot Springs, Va., for the Summer Meeting 1938 and sessions will be held on June 20 to 22. The hotel is situated in a park of several thousand acres at an elevation of 2500 ft and the picturesque surroundings make it ideal for a summer meeting.

There are many ways to enjoy the outdoor diversions at Hot Springs: golf on the Homestead or the Cascade courses, riding and driving on the fine trails and woodland roads, tennis and skeet, swimming in the outdoor pools at the Cascades and at Warm Springs, and baths in the nearby Spa. For those desiring less strenuous sports, there are horseshoe-pitching, croquet, fishing and interesting walks near the hotel as well as indoor diversions such as badminton, ping pong, cards, daily movie shows and concerts.

Pres. E. Holt Gurney has appointed a Committee on Arrangements, the personnel of which is J. A. Donnelly, Largent, W. Va., *General Chairman*, J. K. Peebles, Jr., Charlottesville, Va., John Shanklin, Charleston, W. Va., L. Ourusoff, Washington, D. C., I. B. Helburn, Cincinnati, O., E. W. Klein, Atlanta, Ga., and R. M. Johnston, Blacksburg, Va.

Plans are being formulated for three technical sessions at which 12 papers and reports will be presented, an informal discussion of local chapter problems, meetings of the Council and Nominating Committees, Committee on Research and many other technical committees.

An entertainment program for members and ladies is being planned so that there will be something to do for everybody. The golfers will contend for the Research and Chapter Cups and there will be a banquet and dance.

Service at The Homestead is on the American Plan and rates are $9 per person per day for two people in a room, $10 per person per day for single occupancy.

Accommodations are available for 60 people in Vine Cottage and Cascades Inn where the rate is $6 per day per person American Plan. (Vine Cottage is five minutes walk from The Homestead and Cascades Inn is three miles away).

Good roads lead to The Homestead at Hot Springs from every direction and members from New England, New York, Philadelphia, Baltimore and Washington, will find that the Shenandoah Valley and the Skyline Drive contain much of scenic interest; while those members from Toronto, Buffalo, Pittsburgh and Cleveland, will discover that Keyser Ridge and the Alleghenys provide an interesting route. Members coming from the southeast will pass through country of great historic interest and will enjoy driving through the Smoky Mountain district. Those from the West coming via Charleston will find many interesting places along the route to Covington and Hot Springs.

Hot Springs is reached by the Chesapeake and Ohio Railroad. Trains are air conditioned and overnight service is available from New York, Philadelphia, Washington, Cleveland, Detroit and Cincinnati.

The round trip railroad fares are given in the following table:

City	Round Trip Railroad Fare	One Way Pullman Lower Berth
Atlanta	$ 30.20	$ 5.50
Boston	42.85	4.00
Buffalo	36.70	5.50
Chicago	39.45	5.00
Cleveland	29.45	4.25
Cincinnati	23.05	2.50
Des Moines	52.70	7.50
Detroit	31.90	4.25
Dallas, Tex.	50.65	9.50
Grand Rapids	38.10	5.00
Houston, Tex.	54.20	10.25
Kansas City	50.35	7.50
Los Angeles	122.25	19.75
Milwaukee	44.10	6.50
Minneapolis	57.00	7.50
New York City	27.25	3.00
Oklahoma City	51.60	9.00
Omaha	55.90	8.00
Philadelphia	22.05	3.00
Pittsburgh	31.65	4.00
San Francisco	122.55	19.75
Seattle	123.60	20.00
St. Louis	40.80	5.00
Washington, D. C.	14.05	2.50
Montreal	57.68	6.00
Toronto	46.93	6.00
Winnipeg	77.00	9.00

Hot Springs is located in picturesque mountain country secluded from the outside world and The Homestead is strictly in keeping with its surroundings where simplicity, dignity and charm are nicely blended. It is an ideal place for the Society's Summer Meeting 1938.

Radio Audience Hears About Air

ONE of the interesting highlights at the 44th Annual Meeting of the Society was a broadcast arranged from one of the ventilating towers of the new Lincoln Tunnel in New York.

David Driscoll of the special features division of the Mutual Broadcasting Co., handled the broadcast which went over the entire Mutual net work of 68 stations from coast to coast.

Participating in the broadcast program were A. C. Davis, superintendent of maintenance of the Port of New York Authority projects and a member of the Society, E. Holt Gurney, President of the Society, and Walter L. Fleisher, Chairman of its Committee on Research. Following is a transcript of the broadcast:

DRISCOLL: Good afternoon, ladies and gentlemen, this is Dave Driscoll speaking from the control room of the New York Land Building of the Port Authority at 39th street and 11th Avenue. THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS has been in convention in this city during the entire week and as a salute to this organization and to the progress made in heating and ventilation we are going to take you on a quick radio tour of this giant shaft, one of three, which alternately blows in and sucks out of the Lincoln Tunnel almost 2,000,000 cu ft of air a minute, one of the greatest ventilation feats in history.

The Lincoln Tunnel connects midtown Manhattan with Weehawken, N. J., and a great network of highways that extend to all parts of the United States. It is the newest contribution to fast, efficient and modern motor transportation on the Eastern seaboard, a sister as it were to the already world famous Holland Tunnel at the lower end of the city of New York.

With me in this control room are A. C. Davis, superintendent of maintenance of the Port Authority projects; E. Holt Gurney, newly elected president of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, and Walter L. Fleisher, chairman of the Society's Research Committee. With such guides it should be easy to find our way around the maze of blowers, fans and other equipment in this building.

Mr. Davis, how are these hundreds of thousands of cubic feet of air pumped into the tunnel, and likewise how are they drawn out?

DAVIS: At maximum capacity 1,736,000 cu ft of air are circulated in the Lincoln Tunnel. It is blown into the shaft by great blowers in three separate buildings, of which this is one. This is known as the New York Land Building and is responsible for the air conditioning of the tunnel from the water's edge to the New York portal. About two blocks from here, right on the edge of the Hudson river, another great plant is operated that purifies the air from the shore to the center of the river. On the New Jersey side another plant takes care of the tunnel from the center of the river to the Weehawken portal.

At Lincoln Tunnel Broadcast: (*left to right*) A. C. Davis, David Driscoll, Pres. E. Holt Gurney, and W. L. Fleisher

DRISCOLL: How does the air actually circulate, Mr. Davis?

DAVIS: It is forced from the blowers to the bottom of the tunnel. The roadbed is elevated about 6 ft above the very bottom of the tube; in other words the roadway is really a false bottom and beneath it flows the air which is allowed to filter through ducts along the lower edge of the highway. The roof of the tunnel is of a similar design, except the air is sucked *out* of this cavity, back through great pipes to this building, carried aloft and finally released through the roof.

DRISCOLL: How much power does it take to operate the blowers and exhaust fans?

DAVIS: It takes a lot of power. All in all there are 32 fans. We have 15 huge fans, ranging in horsepower from 17 to 270 hp. Twelve of these fans are located in this building, 7 in the River Building and 13 on the New Jersey side. The capacity of these fans ranges from 10,600 to 280,000 cu ft of air a minute. Their total capacity per *minute* is the equivalent of 64 *tons!* The power of the fans ranges from 17 to 270 hp.

DRISCOLL: I think it might be interesting to mention that this building has six floors, each with its quota of fans and exhausters. In the position in the control room on the ground floor the only sound is that of a deep hum, but up on the other floors it's a different story. Mr. Davis, a microphone has been installed on the floor above where the fans have been shut down. If you will give the order to start things rolling the audience will be transferred to a point within 2 ft of one of them and our friends can hear what it sounds like.

DAVIS: That's a good idea. I'll phone to the main control room of the entire system. That's on the New Jersey side of the river.

The fans will start to roll in just a moment.

DRISCOLL: I have a pair of headphones here that will tell me when they start.

Mr. Gurney and Mr. Fleisher have been standing by with their eyes on the rows of twinkling lights on the control panels, looking like the proverbial cat that ate the mouse. Watching the progress of engineering must be of particular satisfaction to engineers such as you. How about it, Mr. Gurney?

GURNEY: There's only one phrase that describes this plant. It's a honey! Many of the engineers of our Society made trips through here during the past week and they all were of the same opinion.

DRISCOLL: Little does the motorist realize when he speeds through the Lincoln Tunnel what hundreds of thousands of man hours have been put into the construction of devices for his safety and convenience. There go the fans though, so let's switch our controls to the first floor blower room.

DAVIS: Mr. Driscoll, perhaps, you would like to have a look at the room where we test the purity of the tunnel air?

DRISCOLL: Fine, lead the way. Mr. Davis, what was the cost of construction of this plant?

DAVIS: Roughly $2,000,000 for the construction of the towers, the installation of the ventilation fans and the control equipment. That doesn't include the installation of the ducts in the tunnel which were considered part of the general construction.

DRISCOLL: Big figures usually mean a big job. Running the Lincoln Tunnel must be quite a task.

DAVIS: Well, figures are sometimes boring but here are a few. The tunnel is 8215 ft from portal to portal, a little under two miles. The lowest point of the roadway is 97 ft below the surface of the Hudson river. Operation of the tunnel requires a

private police force of 113 men and a maintenance crew of 87. Nine hundred seventy 150-watt electric lights line the tunnel walls in addition to the red, green and amber signal lights every 120 ft on either side. Traffic flows in both directions over a roadway 21½ ft wide. When the second tube is completed traffic will operate one way in each, as in the Holland Tunnel.

DRISCOLL: That's enough, Mr. Davis. I can see what a job it is.

DAVIS: All right, but here is the room in which we test the purity of the tunnel air. You will notice all of these chromium tanks and glass tubes, etc.

DRISCOLL: I get the general idea, Mr. Davis, but I'm not going to worry about it. It's a little technical for me. I'll just drive through the tunnel knowing that its operation is in capable hands. It's some contrast to the days when we had to wait in line, sometimes for hours, for a ferryboat. Now we are whisked beneath the river in less than 4 min. But let's turn to Mr. Gurney and Mr. Fleisher for a few moments. Mr. Gurney, as President of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, can you tell us what a project such as we have seen today means to the layman?

GURNEY: This is another wonderful example of man's triumph over the forces of nature. What is being done in this tunnel is, in effect, to harness the air and to create and control a man-made whirlwind that thousands of autoists may travel under the Hudson River in comfort.

Indeed, it is more important than this, for such tremendous engineering feats as the Lincoln Tunnel would have no commercial usefulness whatever were it not for the science of ventilation. When the Egyptians, 2000 years and more ago, invented crude devices by which huge fans of ostrich feathers were manipulated by slaves to make their monarchs comfortable, they were engaging in the science of ventilation in its most primitive form.

Today we have efficient, silent electric fans, blowers to distribute air evenly and effectively throughout our homes, and have developed out of early knowledge about ventilation a whole new science which is called air conditioning.

The AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS is happy to take part in this instructive broadcast because for 20 years it has devoted itself to the development and application of new knowledge to ventilation in the public interest.

DRISCOLL: And now Mr. Fleisher, what significance do you see in this great engineering project?

FLEISHER: It seems to me that there is a definite relationship between this highly interesting and spectacular ventilating system and the every-day experiences of people in their homes.

It illustrates one of the cardinal principles of air conditioning in that air movement and air purity are as important to our comfort as is the correct temperature and humidity of the air we breathe.

What is being done here on a large scale is similar in principle to what the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, through its cooperative and individual research, is doing to improve technical knowledge about home ventilation and all the factors which enable engineers to create an artificial condition of the atmosphere most conducive to health and comfort.

In the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS at Pittsburgh, we are studying many different phases of ventilation both from the standpoint of health and disease—from finding out about drafts to investigating the effect of wind velocity against tall buildings. In fact, ventilation is coming to play an increasingly important part in air conditioning.

Just as it keeps the air pure in the Lincoln Tunnel, so in our homes it drives out stale air, helps to dissipate odors and remove excess body heat. No more striking example of its value may be found than in this very interesting application which we have had the privilege of seeing and interpreting to the radio audience.

W. A. Russell Joins Hoffman Specialty Co.

A recent announcement from Hoffman Specialty Co., Waterbury, Conn., confirms the appointment of W. A. Russell to a responsible position with the General Sales Department. Mr. Russell will have his headquarters at the New York office and

W. A. Russell

his work will consist chiefly of promoting the distribution and sale of certain controls and heating specialties which have been developed by the organization.

Mr. Russell comes to the Hoffman Specialty Co. with a wide acquaintance and seasoned experience gained during his 20 years with the United States Radiator Corp. in executive capacities at Kansas City, Chicago and Detroit.

Prof. Philip Drinker Addresses Ontario Chapter

February 7, 1938. The regular meeting of the Ontario Chapter was held at the Royal York Hotel, Toronto, with 53 members and guests present and Vice-President Jenney presided in the absence of Pres. G. A. Playfair.

E. Holt Gurney, president of the A.S.H.V.E., spoke briefly to his fellow chapter members on the Society's Annual Meeting which was held in New York.

Dr. Cunningham introduced the principal speaker of the evening, Prof. Philip Drinker of the Harvard School of Public Health, Boston, Mass. Professor Drinker spoke on Air Pollution and Dust Control and a very interesting discussion followed his address. M. F. Thomas expressed a vote of thanks on behalf of the chapter to Professor Drinker for his splendid talk.

Thomas McDonald presented H. R. Roth with a traveling bag as a gift from the Chapter, in appreciation of the work which he has done as secretary of the Ontario Chapter for the past several years.

Michigan Chapter Enjoys Dinner Dance

February 12, 1938. The Michigan Chapter of the A. S. H. V. E. held their annual Valentine dinner dance at Dearborn Inn. There were 152 present including members, guests and friends, according to the report of Secy. G. H. Tuttle. Music was furnished by Dave Diamond and his orchestra, and the party ended at 1:00 o'clock, everyone having a delightful evening.

Elected To Life Membership

R. H. Feltwell
Washington, D. C.
Elected **1905**

Walter Leek
Vancouver, B. C.
Elected **1903**

T. N. Thomson
Huntington, N. Y.
Elected **1899**

Washington, D. C. Chapter Holds Regular Meetings

February 2, 1938. Secy. L. F. Nordine has reported that the routine business was dispensed with at this meeting of Washington, D. C. Chapter, and the speaker of the evening was William Goodman, research engineer, Trane Co.

Mr. Goodman's subject was Dehumidification of Air with Coils and in his talk he thoroughly covered air conditioning problems.

The meeting was an enthusiastic one and approximately 60 members and guests attended.

According to Secretary Nordine the financial condition of the Chapter is satisfactory and its strength in membership and finance is constantly growing.

January 12, 1938. Fifty members and guests of Washington, D. C. Chapter were present and at this meeting 18 new members affiliated with the local organization.

Pres. L. Ourusoff presided and announced continued activity with the Department of Health on cooperative rules and standards for improving air conditioning and heating installations in the District of Columbia. The activity of the Chapter with reference to the registration of engineers for the District of Columbia was brought to the attention of the meeting.

Following the business session Mr. Kugle, smoke inspector, District of Columbia, was the speaker of the evening. In his talk he outlined the history of smoke inspection, local smoke regulations from an engineering viewpoint, problems with reference to health and economy, and local policies with reference to the owner, architect, designer, contractor and engineer.

December 8, 1937. An interesting meeting of Washington, D. C. Chapter was preceded by dinner with about 50 members and guests attending.

Pres. L. Ourusoff presided and the principal speaker was F. E. Spurney, whose subject was Mechanical Construction and Installation Problems in Modern Building.

Mr. Spurney's talk concerned labor unions, cooperation between engineers, contractors and architects, and problems of building construction.

November 10, 1937. A dinner was held prior to the regular meeting of Washington, D. C. Chapter, and as there was no particular business to transact, the meeting was turned over to Prof. C. P. Yaglou, Harvard School of Public Health, Boston, Mass., who was the speaker.

An address on Air Conditioning and Its Effect on Health, Comfort and Disease was given by Professor Yaglou, and was very interesting to the 55 members and guests present.

October 13, 1937. Approximately 40 members and guests attended the meeting of Washington, D. C. Chapter, when the business meeting was preceded by the usual dinner and social hour.

The first order of business was Pres. L. Ourusoff's report of activities of the officers during the vacation period, and an announcement of plans for the 1937-38 season.

C. W. Walton, Jr., mechanical engineer, Rockefeller Center, New York City, addressed the meeting and described the operation and maintenance of the air conditioning systems at Rockefeller Center.

Condensation in Building Walls Interests Wisconsin Chapter

January 17, 1938. Members of the Wisconsin Chapter held a meeting at the City Club, Milwaukee. The speaker of the evening was L. V. Teesdale, senior engineer, Forest Products Laboratory, Madison, Wis.

C. H. Randolph introduced Mr. Teesdale, who spoke on Condensation in Building Walls. Mr. Teesdale stated that condensation difficulties in this country are common in the region north of the Ohio River and east of the Rockies. In the address, data were presented which were obtained from a test house on the roof of the laboratory. This house was constructed with various wall sections. Additional data were obtained from test rooms in the laboratory, in which a fixed condition, zero temperature, and 10 per cent relative humidity was maintained.

In general Mr. Teesdale recommended as a means of protection against moisture the use of a vapor barrier, such as paper under the plaster base where homes are under construction, and aluminum paint on the plaster for existing homes.

The questions which followed the formal presentation gave evidence of the interest of Wisconsin Chapter in the subject.

December 20, 1937. There were 28 members and guests at a dinner of Wisconsin Chapter held at the City Club, Milwaukee, and several additional members came later for the coffee talk and meeting, according to the report of Secy. H. C. Frentzel.

Prof. G. L. Larson showed a movie taken during a fishing trip through upper Ontario. These movies were very interesting from the fact that they were taken in primeval country, together with Professor Larson's description of their living conditions and their sole dependence upon their own resources. Furthermore any question as to the size of the fish and the quality of the fishing country was eliminated.

The coffee talk was followed by an open forum in which pressure drops through ventilating hoods in industrial application, as given in the Wisconsin Safety Code, were discussed. A motion was made and seconded that the Board of Governors write the Industrial Commission of Madison with reference to this portion of the code.

Other subjects under discussion were ventilation of auditoriums and the question as to whether or not boilers should be installed for a total maximum load or only for alternate loading.

After discussing the possibility of changing future nominations for the Board of Governors, it was voted that the Board of Governors prepare a suitable amendment to the Constitution to be submitted to the Chapter for approval.

Texas Chapter Holds Meeting in San Antonio

January 15, 1938. Texas Chapter held a meeting at the St. Anthony Hotel in San Antonio, when 49 members and guests gathered from Houston, Dallas, College Station, Kingsville, Austin, and San Antonio.

Pres. R. F. Taylor presided, and following dinner he called upon M. L. Diver and A. J. Rummel to present the guest speakers of the evening. W. D. Masterson, manager, City of San Antonio Water Board, presented a paper on The Relation of Air Conditioning in Texas to the Water Supply; Prof. H. E. Degler, Department of Mechanical Engineering, University of Texas, spoke on the subject of The Place of Ice in Air Conditioning.

Secy. W. H. Badgett has reported that the Chapter went on record as heartily endorsing the effort of the Kansas City Chapter to secure the 1940 Annual Meeting of the Society and Exposition, or the 1941 Annual Meeting.

It was reported by Secretary Badgett that there were 30 applications for membership in the Society from Texas pending action of the Council, and that upon their election there would be 82 members in the state, an increase of 100 per cent since the Chapter was organized a year ago.

The meeting adjourned to meet in Houston February 12.

44th Annual Meeting Reviewed to Atlanta Chapter

February 1, 1938. Pres. E. W. Klein presided at the regular meeting of the Atlanta Chapter which was held at the Biltmore Hotel.

After the reading of the minutes of the previous meeting, which were approved, and the reading of several communications

received from A.S.H.V.E. headquarters, President Klein gave a complete and comprehensive report of his trip as Chapter Delegate to the Annual Meeting of the Society and the 5th International Heating and Ventilating Exposition. Following this, Lawrence Kent gave an interesting discussion on some of the papers which were presented at the meeting of the *National Warm Air Heating and Air Conditioning Association* which was held in New York during the time of the A.S.H.V.E. meeting.

Discussion of future meeting programs followed and H. K. McCain was appointed to communicate with Albert Buenger, Dayton, Ohio. It was also decided to contact Herman Seid to speak to the Chapter at some future date on the subject of Air Conditioning Patents. M. G. Driscoll will address the Chapter at an early date on Silica Gel.

According to the report of Secy. C. T. Baker the meeting adjourned at 9:30 p. m.

St. Louis Chapter Has C. S. Leopold and Dr. Mills as Speakers

January 27, 1938. St. Louis members of the A.S.H.V.E., the *A.I.A.*, and the Engineers Club held a joint meeting at the Engineers Club when 220 were present.

C. S. Leopold, consulting engineer, Philadelphia, was introduced by J. H. Carter, and spoke on Air Conditioning Problems in tonnages of 100 to 1200 tons.

Preceding the meeting at the Engineers Club, a dinner was held at the Coronado Hotel for Mr. Leopold at which 15 members and guests of the A.S.H.V.E. Chapter were present.

January 4, 1938. Vice-Pres. E. E. Carlson called to order the meeting of St. Louis Chapter at Garavelli's restaurant with 20 members and 19 guests present.

The routine business included the reading of the minutes of the previous meeting, the treasurer's report, and reports of standing committees.

After a short recess, Mr. Carlson introduced the guest speaker, Dr. C. A. Mills, professor of Experimental Medicine, University of Cincinnati. Dr. Mills presented a paper on Physiologic and Health Factors as Related to Air Conditioning and illustrated his talk with slides. Dr. Mills' paper was well received as was evidenced by the questions and discussion which followed.

Effect of Air Conditioning on Disease Subject of Iowa-Nebraska Chapter

February 8, 1938. The sixth meeting of the Iowa-Nebraska Chapter was held at the Hotel Kirkwood, Des Moines, Ia., with 28 present.

Vice Pres. A. L. Walters presided, and after the reading of the minutes, which were approved, he gave a detailed report of the Society's Annual Meeting held in New York City during the week of January 24.

Secy. W. R. White suggested that the next Chapter meeting be held at the University of Nebraska, to be followed by a meeting at the Iowa State College, thus holding one meeting a year at each of these two colleges.

At the conclusion of the business meeting Vice President Walters introduced the speaker of the evening, Dr. E. Vernon Hill, Chicago, Ill., who spoke on the subject of The Effect of Air Conditioning on Comfort, Health and Disease. According to Dr. Hill, many diseases are contracted through the nose and mouth. He pointed out the advance that has been made by the medical profession in the elimination of certain diseases, and showed that sanitation has been responsible for a drastic decrease in typhoid and similar diseases. On the other hand, he related that there had been almost no progress shown in combatting the diseases associated with the respiratory organs. Dr. Hill illustrated his lecture with slides.

A discussion followed Dr. Hill's very interesting talk, including hospital air conditioning, and Mr. Walters expressed the appreciation of the Iowa-Nebraska Chapter in having Dr. Hill with them. The meeting adjourned at 10:30 p. m.

Heat Control and Insulation Discussed at Philadelphia Meeting

February 10, 1938. The regular meeting of the Philadelphia Chapter was held at the Engineers Club and the routine business included the reading of the minutes of the previous meeting and the treasurer's account, which were approved.

Reports covering the activities of the Membership and the Meetings Committees were presented, and R. C. Bolsinger spoke on the Society's Annual Meeting held in New York during January.

A plea in behalf of the campaign for funds being conducted by the Salvation Army was made by Gus Spahr, Philadelphia Electric Co.

Secy. C. B. Eastman received the following obituary of G. F. Stone, which was prepared by J. D. Cassell and R. C. Bolsinger.

George F. Stone, Life Member of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, was born August 22, 1854, at Boston, Mass., where he attended the public schools which was the extent of his education, other than personal study and application to the requirements of his chosen profession, which he acquired thoroughly and well, as attested by the following record: 4 years with A. S. Jones, Philadelphia, Pa., as engineer and superintendent; 4 years with G. S. Stone, Bristol, Pa., as engineer and superintendent; 4 years with Chas. Onderdouk & Co., Philadelphia, as engineer, estimator and superintendent; 4 years with H. M. Sciple & Co., Philadelphia, as engineer, estimator and superintendent; 3 years with Clarkson, Scott & Co., Philadelphia, as engineer, estimator and superintendent; 4 years with A. A. Griffin Iron Works, Jersey City, N. J., as engineer, estimator and superintendent; 18 years with Chas. E. Monday & Co., Philadelphia, as engineer, estimator and superintendent.

Mr. Stone started his career at a time when one had to depend on his own resources to accomplish his ends, as most men engaged in the Art did not have the benefits of technical education and research so readily at hand today.

George F. Stone joined the A.S.H.V.E. and was made a Life Member in 1930; he was also a member of the Philadelphia Chapter of which he was a constant attendant. His was a pleasing and helpful personality to all who knew him.

He died July 16, 1937 in his 83rd year and was active until within a few months of his death. His widow survives him.

An interesting moving picture on the subject of Heat and Its Control was presented. Following the picture, talks were given by N. A. Humphrey and L. C. Leslie, Johns-Manville Co., on Selection of the Economic Thickness of Insulation.

Death of W. E. Kemery, Jr.

The Society lost one of its new members in Portland, Oregon, in the death of William E. Kemery, Jr., which occurred on January 8 after a short illness. Mr. Kemery who was 35 years old was a member of the architectural firm of A. E. Doyle & Associate.

Born at Parkersburg, W. Va., Mr. Kemery attended the public schools there and later was graduated from Carnegie Institute of Technology, Pittsburgh. He was successively associated with two firms there, but came to Portland soon after his graduation. With the architectural firm in Portland he was chief engineer in charge of mechanical construction. He had charge of the construction of the J. P. Finley mortuary and chapel, the new Portland Art Museum, the Willamette University Library, the remodeling of the Portland Union Station, the building of the McMinnville Public Library, as well as branch banks and other buildings.

The Officers and Council of the A.S.H.V.E. extend their sympathy to Mr. Kemery's survivors, who include his widow, Mrs. Charlotte Kemery, of Portland, his mother, Mrs. V. L. Kemery and a brother, Robert, of Parkersburg, and a sister, Mrs. William Bolton, of Charleston.

CANDIDATES FOR MEMBERSHIP

The Constitution of the Society, as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the names of applicants and their references shall be printed in the next issue of the JOURNAL of the Society or sent to the members in other approved manner as ordered by the Council. When replies are received from references, the Candidate's application shall be submitted to and acted upon by the Committee on Admission and Advancement as soon as possible.

When the Committee on Admission and Advancement has acted favorably upon a Candidate's application and assigned his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month 36 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Committee on Admission and Advancement, and in turn, the Council, urge the members to assume their share of responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned.

All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some member by April 15, 1938, these candidates will be balloted upon by the Council. Those elected to membership will be notified by the Secretary immediately after election.

CANDIDATES	REFERENCES	
	Proposers	*Seconders*
ABRAMSON, RALPH, Htg. & Air Tempering Engr., Max Miller, Chicago, Ill.	Max Miller (*Non-Member*) F. L. Hutchinson (*Non-Member*)	Thomas Wilson (*A.I.E.E.*) B. M. Garland (*Non-Member*)
BOEHM, H. F., JR., Exec. Vice Pres., Beacon Air Cond. & Controls Corp., Beacon, N. Y.	A. L. Baum Harry Fiedler	A. C. Buensod Gerald Rose (*Non-Member*)
CAMERON, R. T., Student, Carnegie Inst. of Tech., Pittsburgh, Pa.	C. M. Humphreys T. F. Rockwell	P. A. Edwards R. B. Stanger
CAPLE, IRA, Student, University of Minn., Minneapolis, Minn.	F. B. Rowley A. B. Algren	R. C. Jordan H. M. Betts
CLARK, L. W., Engr., Hall-Neal Furnace Co., Indianapolis, Ind.	G. A. Voorhees J. D. Hoffman	E. S. Hildreth S. E. Fenstermaker
DOERR, C. F., Student, Lewis Inst., Chicago, Ill.	V. L. Sherman J. S. Kozacker (*A.S.M.E.*)	J. G. Bennett (*A.S.C.E.*) F A. Rogers (*A.I.E.E*)
DUTCHER, H. S., Engr., A. L. Hart, Inc., Brooklyn, N. Y.	Bernard Leventhal Irwin Jalonack	Henry Gitterman J. R. Potter
DYKMAN, J. G., Htg. Contractor, Grand Rapids, Mich.	W. W. Bradfield H. D. Bratt	F. C. Warren C. H. Morton
EISELE, D. E., Owner, Eisele Engrg. Co., Appleton, Wis.	W. G. Boales E. N. McDonnell	T. A. Crawford (*Non-Member*) J. F. Tanton (*Non-Member*)
GRAY, E. W., Air Cond., Commercial Dept., Oklahoma Gas & Elec. Co., Oklahoma City, Okla. (*Advancement*)	E. F. Dawson R. G. Dolan	F. E. Giesecke W. A. Russell
GUSTAFSON, C. A., Sales Engr., The Powers Regulator Co., Chicago, Ill.	M. W. Bishop M. J. Bamond	A. L. Crump Tom Brown
HENDERSON, A. S., Jr. Engr., Unit Air Conditioners Pty., Ltd., Sydney, Australia	J. W. Picot J. A. Robinson	Arthur Pilz (*A.M.I.E.*) J. H. Liddle (*A.S.R.E.*)
HINNANT, C. H., JR., Chief Engr., Hungerford Coal Co., Richmond, Va.	H. M. Rudio J. A. Johnston	W. G. Stephenson (*Non-Member*) E. T. Nichols (*Non-Member*)
HOFFBERGER, J. P., Sales Mgr., Wagner Elec. Corp., St. Louis, Mo.	G. W. F. Myers C. V. Allen	L. R. Szombathy G. B. Trahn (*Non-Member*)
HOLLAND, G. R., Asst. Chemist, American Sugar Ref. Co., New York, N. Y.	J. F. Kern, Jr. Lewis Graves (*Non-Member*)	W. J. Morf (*Non-Member*) H. J. Bastone (*A.C.S.*)
HOMAN, J. J. D., Engr., Tampa Armature Works, Inc., Tampa, Fla.	H. C. Flanagan (*Non-Member*) J. C. Moxley (*Non-Member*)	Albert Haworth (*Non-Member*) R. A. Menendez (*A.I.E.E.*)
JAKOBY, A. C., Estimator & Designer, Sears Roebuck & Co., Philadelphia, Pa.	Mitchel Landau C. W. Whitney	M. F. Blankin H. H. Erickson
KELLER, G. A., Asst. to Supt. of Engrg. & Maint., Abraham & Straus, Brooklyn, N. Y.	C. R. Hiers J. W. Hunter (*A.S.M.E.*)	R. B. Dale (*A.S.M.E.*) A. S. Egilsrud (*A.S.M.E.*)
KUCHER, A. A., Mgr. Air Cond. Engrg., Frigidaire Div., General Motors Corp., Dayton, Ohio	H. B. Hull Rothwell Woodward	Albert Buenger R. E. Gould (*Non-Member*)
KUNEN, HERBERT, Mech. Engr., Anemostat Corp. of America, New York, N. Y.	F. J. Kurth Fritz Honerkamp	Ladislav Fischer C. R. Hutcheon
LANG, JACOB, Proprietor, Lang & Lang, Richmond Hill, L. I., N. Y.	F. B. Herty W. H. Ballman	J. J. Deely J. E. Maynard
LEIGH, R. L., Engr., Hart & Cooley Mfg. Co., Holland, Mich. (*Reinstatement*)	S. Konzo S. H. Downs	J. H. Walker A. E. Knapp
ROBINSON, E. R., Development Engr., May Oil Burner Corp., Baltimore, Md.	J. deB. Shepard E. L. Crosby	J. E. Seiter R. L. Leilich
ROSENBLATT, A. M., Partner, Rosenblatt & Hunt, Charleston, W. Va.	V. S. Day W. H. Driscoll	W. H. Carrier Donald French

CANDIDATES	REFERENCES	
	Proposers	*Seconders*
RYAN, J. B., Chief Engr. & Mgr., Engrg. Service Co., Kansas City, Mo.	L. A. Stephenson David Caleb	Carl Clegg K. M. Stevens
SMALL, B. R, Managing Engr., Guilford Engrg. Co., Greensboro, N. C. (*Advancement*)	H. P. Gant W. A. Bornemann	C. L. Templin N. E. Sheldon
SPENCER, W. E., Br. Mgr., National Radiator Corp., Buffalo, N. Y.	B. C. Candee M. C. Beman	L. A. Harding Joseph Davis
STANLEY, R. L., Design Engr., Payne Furnace & Supply Co., Inc., Beverly Hills, Calif.	Leo Hungerford E. H. Kendall	A. J. Hess Harrold English
TEZOS, O. A., Asst. Ind. Gas Engr., Consumers Power Co., Pontiac, Mich.	D. H. Gerhard J. S. Kilner	B. G. Campbell (*Non-Member*) H. C. Haroldson (*Non-Member*)
TURK, LEONARD, Student, Carnegie Inst. of Tech., Pittsburgh, Pa.	C. M. Humphreys T. F. Rockwell	P. A. Edwards R. B. Stanger
WADDINGTON, B. C., Chief Engr., Major Appliance Co., Omaha, Nebr. (*Reinstatement*)	W. R. White Albert Buenger	Oliver Crose (*Non-Member*) H. S. Seymour (*Non-Member*)
WALTHER, F. G., Drafting & Engrg., York Ice Mchy. Corp., Brooklyn, N. Y.	J. G. Eadie E. E. Ashley	H. H. Bond L. F. Bourgarde (*A.S.R.E.*)
WHITE, E. G., Asst. Custodian Engr., Bronx Central Post Office, Bronx, N. Y.	F. T. Bankauf (*Non-Member*) T. C. Randall (*Non-Member*)	L. P. Gollerstepper (*Non-Member*) J. E. Mulberry (*Non-Member*)
WONSON, A. S., JR., Student, Beverly Trade School, Beverly, Mass.	P. A. L. Foulds N. J. H. Shaw	H. B. Wiegner J. W. Brinton
WOODS, E. H., Mgr., Higgins & Zabriskie, Ithaca, N. Y. (*Reinstatement*)	W. M. Sawdon A. A. Barns	W. H. Elwood J. W. Williams
WORMLEY, R. F., Br. Mgr., Grinnell Co. of Can., Montreal, Que., Can.	G. L. Wiggs W. W. Timmins	J. R. Dupuis C. W. Johnson

Candidates Elected

In the past issues of the JOURNAL of the Society the names of the following men were listed as Candidates for Membership. The membership grade of each Candidate has been assigned by the Committee on Admission and Advancement and balloted upon by the Council. We are now instructed by the Council to post herewith, as required by Art. B-III, Sec. 8, of the By-Laws, the following list of candidates elected:

MEMBERS

ADDINGTON, H. B., Consulting Engr., Brooklyn, N. Y.

ALFREY, H. F., Chief Engr., Milwaukee Gas Specialty Co., Milwaukee, Wis.

BAKER, W. C., Air Cond. Engr., Frigidaire Div., Nashville, Tenn.

BIGELOW, E. S., Mgr. Air Cond. Div., Trilling & Montague, Philadelphia, Pa.

CALL, JOSEPH, Air Cond. Engr., Elliott-Lewis Co., Philadelphia Pa. (*Advancement*)

CARROLL, A. F., Development Engr., Automatic Prod. Co., Milwaukee, Wis.

CHALMERS, C. H., Owner & Gen. Mgr., Chalmers Oil Burner Co., Minneapolis, Minn. (*Reinstatement*)

CHASE, L. R., Dist. Supervisor, The Carters-Waters Corp., Kansas City, Mo. (*Advancement*)

CHENEVERT, J. G., Consulting Engr., Arthur Surveyer & Co., Montreal, Que., Can.

CHILDS, L. A., Dist. Sales Mgr., Clarage Fan Co., Philadelphia, Pa.

COOPER, D. S., Air Cond. Engr., Air Conditioning Co., Houston, Texas (*Advancement*)

FERDERBER, M. B., Dept. of Ind. Hygiene, Univ. of Pitt. Medical School, Pittsburgh, Pa.

GRAY, J. W., Member of Firm, The Gray-Henry Co., Bay City, Mich.

GREENWOOD, O. J., Air Cond. Engr., Walgreen Co., Chicago, Ill.

HALL, W. L., Engr., Frank A. Leon Co., Washington, D. C.

HARDEN, J. C., Htg. Engr., Round Oak Co., Dowagiac, Mich.

HAZLETT, T. L., Medical Director, Westinghouse Elec. & Mfg. Co., East Pittsburgh, Pa.

HUBER, ENRIQUE, Mgr. of Testamentaria, de E. Huber, Ing., Mexico, D. F.

JOHNSON, T. R., Br. Mgr., The Trane Co., Des Moines, Iowa (*Reinstatement*)

KRUSE, W. C., JR., Owner, Kruse Engrg. Co., Newark, N. J.

LANDES, B. E., Consulting & Design Engr., Des Moines, Iowa

MACEACHIN, G. C., Dist. Engr., Frigidaire Div., Ft. Worth, Texas

MARZORATI, GIUSEPPE, Consulting Engr., Societa Giacomo Jucker, Milano, Italy

MCCAFFRAY, C. E., Design Engr., Henry Adams, Inc., Baltimore, Md.

MCCARTHY, T. F., Div. Mgr., Frigidaire Corp., Chicago, Ill.

MEYER, K. A., Design Engr., L. J. Mueller Furnace Co., Milwaukee, Wis.

MILLER, D. C., Office of Alfred Kellogg, Cons. Engr., Boston, Mass.

MOREHOUSE, J. S., Prof. Mech. Engrg., Villanova College, Villanova, Pa.

NELSON, R. O., Htg. & Air Cond. Engr., Fedders Mfg. Co., Chicago, Ill.

OELGOETZ, J. F., Owner, J. F. Oelgoetz Co., Columbus, Ohio

PARVIS, R. S., Engr., Diamond Ice & Coal Co., Wilmington, Dela.

PLANT, E. B., Asst. Engr., Can. Pacific Ry. Co., Montreal, Que., Can.

SPREKELMEYER, J. M., Mgr., General Engrg. Corp., Ft. Worth, Texas

STAHL, W. A., Operating Engr., The Merchandise Mart, Chicago, Ill.

STENCEL, R. A., Chief Engr., Can. Ice Mach. Co., Ltd., Toronto, Ont., Can.

STORCH, C. A., Sales Engr., Johnson Service Co., Chicago, Ill. (*Reinstatement*)

THULMAN, R. K., Mech. Engr., Federal Housing Administration, Washington, D. C.

WAGNER, E. K., Sales Engr., The Powers Regulator Co., Philadelphia, Pa.

WASSON, R. A., Eastern Dist. Mgr., Clarage Fan Co., Brooklyn, N. Y.

WILDER, H. P., Chief Engr., The Patterson-Kelley Co., Inc., New York, N. Y.

WOODWORTH, C. Z., Chief Engr., Howard E. Melton, Inc., Oklahoma City, Okla.

YARBOROUGH, T. R., Dist. Sales Engr., Frigidaire Div., Atlanta, Ga.